SEPTEMBER 5, 1911.

SEP 7 1911

POWER

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Contents 377. Advertising Index 66

DUXBAK BELTING

DUXBAK BELTING

Duxbak Waterproof Leather Belting Was Chosen To Replace All The Belts Burned At This Fire

A large manufacturing plant was destroyed by fire, the engine and boiler room alone escaping.

When the plant was rebuilt and equipped they chose Schieren's Duxbak Waterproof Leather Belting to replace the miscellaneous collection of good, bad and indifferent belts they had before.

The new plant has been running over a year now with a coal bill considerably less than before.

There is very little time lost through broken belts and a great deal gained on the work because of improved belt conditions.

The superintendent told our salesman the other day, in a burst of confidence, that he now believed in using *one kind of belting*, THE BEST, Schieren's "Duxbak."

Don't wait till some misfortune compels the purchase of new belts throughout. Replace any unsatisfactory belt with a "Duxbak," and finally you'll have all "Duxbaks."

Send for Price List.

DUXBAK BELTING TRADE MARK

Chas. A. Schieren Company ESTABLISHED 1868

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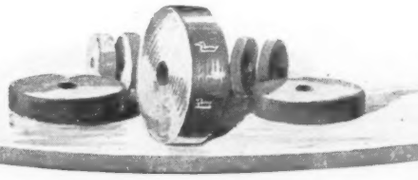
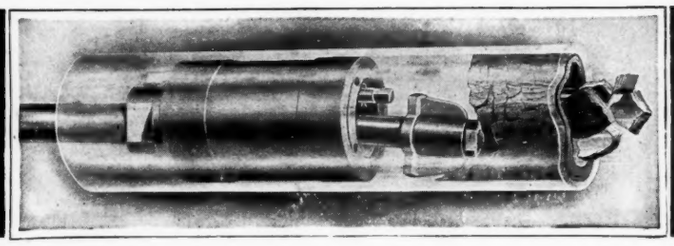
Oak Leather Tanneries, Bristol, Tenn.





The Dean Removing Scale from the Tube of a Water Tube Boiler

Indisputable Evidence

Of Work Done

Wouldn't you like to have some tangible evidence that your scale is removed—
Indisputable evidence that your boilers are clean—
Absolutely clean, not merely less dirty?
Guess work is mighty risky business.
Yet that's all the compound user can do.
He dopes his boilers with compounds.
Then he guesses it's doing the work.

He sees enough scale come out of the boiler at the wash-out to keep up his faith in compounds.

BUT HE DOES NOT AND CAN NOT SEE ENOUGH OF THE INTERIOR OF THE BOILER TO KNOW WHETHER ALL THE SCALE IS REMOVED.

So he guesses—and scale goes on with its damaging work.

The Dean Boiler Tube Cleaner

goes where no inspector can.

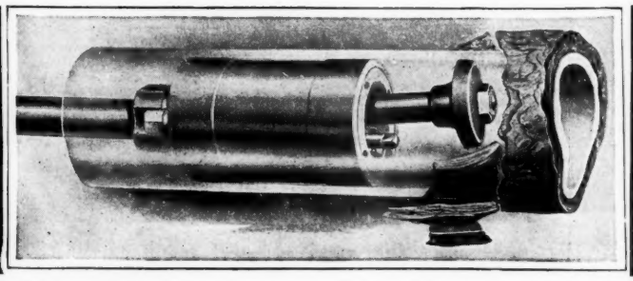
It leaves every tube through which it travels as clean as a gun barrel.
Because it removes scale by the logical way—vibration.
The Dean will give you indisputable evidence that it has cleaned your tubes.
Why fool with guess work when you can be certain?
Try a Dean in one of your boilers, free of charge.
Then contrast old and new results.

You'll see where guess work cost you dearly and where knowledge means money.

Shall we send you a Dean on trial
If you want particulars first, write for Booklet 130.

The Wm. B. Pierce Company

Chicago Office: 801 Steinway Bldg. **Jewett Building, Buffalo, N. Y.**



The Dean Removing Scale from the Tube of a Return Tubular Boiler

KEYSTONE GREASE

KEYSTONE
TRADE MARK REGISTERED U.S. PAT. OFF.

Free

An Engineer's Cap, Brass Grease Cup And Can Of Keystone Grease

We want to send them to you to prove that Keystone Grease is the best, the most economical Lubricant you have ever used on your engines or machinery. To show you how Keystone Grease is wasteless, how it reduces friction to a minimum, will not gum, contains no tallow, fats, soapstone or any injurious substances and is not soluble in water.

Keystone Grease is applied to bearings by pressure grease cups. The development of frictional heat is not required to cause the grease to feed. The original density of Keystone Grease never changes under the severest conditions of speed pressure and atmospheric temperature.

Keystone Grease holds the world's record for the lowest coefficient of friction. We guarantee that no other lubricant on the market can compare with it in quality.

Remember—the coupon brings all three, Cap, Cup and Can of Grease.

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21st, Clearfield and Lippincott Streets Philadelphia, Pa.

Branch offices and warehouses in the principal cities of the United States.
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A SAMPLE FREE

KEYSTONE LUBRICATING CO.,

Please send us a large size sample of Keystone Grease, sufficient for testing purposes, and one Brass Grease Cup and Free Engineer's Cap. It is thoroughly understood that no charges whatsoever will be made for sample, cup or cap, and all express charges will be paid by you.

Name of bearing where sample will be tested . . . . . .
Its width . . . . . Combined horse power of engines . . . . . .
Tap size of cup . . . . . .
FIRM . . . . . .
Per . . . . . .
City . . . . . State . . . . . .
Size cap required. . . . . **CUT OUT THIS COUPON AND MAIL IT TO-DAY** Dept. B—9-5-11

Brass, Top Outlet, Encased Spring

The Object

of a safety valve is to **Relieve an Excess of Pressure, but No More,** and one that does relieve more is wasteful and expensive. Neither is a valve with excessive lift needed, nor is it desirable, because it is not durable, as it causes a tremendous wear on the seating surfaces.

Iron, Encased Spring

Lunkenheimer Pop-Safety Valves

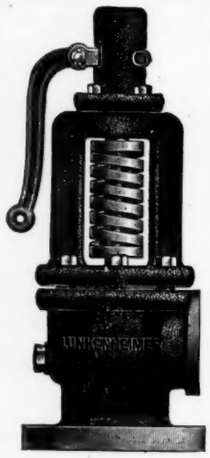
will **promptly Relieve the Overpressure Only,** and nothing more. They can be regulated to reduce the pressure only one pound, if such close regulation is desired; will not chatter or hammer themselves to pieces; will not stick, and adjustment of the pop and pressure can be made from the outside of the valve.

The valves can be had with either brass or iron bodies, made in all standard sizes.

Lunkenheimer Line Of Pop-Safety Valves

is very large and complete one and is thoroughly illustrated and described in our catalogue, a copy of which will be sent free upon request.

"MOST supply houses sell them—yours CAN—if they DONT or WONT—tell US."



Iron, Outside Spring

The Lunkenheimer Company

Largest Manufacturers Of High-Grade Engineering Specialties In The World

General Offices and Works

Cincinnati, Ohio, U. S. A.

New York	Boston	Chicago	London, S. E.
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Brass, Outside Spring

POWER

Vol. 34 NEW YORK, SEPTEMBER 5, 1911 No. 10

THE state of exaggeration will ever try to beat the state of real achievement. It has been at it for centuries, but always loses out. Exaggeration draws attention for a time, but its life is limited; the bigger the boast, the smaller the assurance of fact, and, directly, the shorter its existence.

Exaggeration in a marked degree becomes a parody, a parody of an imaginary original. Just think this out, and the absurdity will be evident. It is like setting up an engine without a foundation.

This state is always found in the fellow who likes to "put it over" the other chap; who believes he is "it" in his own mind and wants to thoroughly impress you with the big "I am." You have met him; does he succeed?

Real achievement is scarce, because it takes a real man's work to get there. It is founded on bedrock, it needs no introduction, it demands no boost or assistance. The matter is in actual evidence, its life is unlimited. A real achievement endures perpetually.

In our little field in this world, the power plant, the ratio of the achiever to the exaggerator is one to a hundred. Are you the one?

Even with this advantage in number of followers, the old "brago" falls behind in the race. He has no firm footing on which to rest. The one is like an Oklahoma cyclone —it sweeps everything in its path; the hundred seek the "cellar," the "'fraid hole"; they hide.

The state of exaggeration in the engineer is the state of self-conceit, and self-conceit has an established record for costing many a man his job. Even after the "jolt" he wonders why; he would not if he could forget the "ego."

The persistent exaggerator is closely allied to the persistent prevaricator—he tells the same yarn so often, and adding each time, he gets to really believe it himself. He always deals with past performances, and is strong on anecdotes with the pronoun "I."

Get him down to facts, to present-day realities. There's a difference. Speak, for instance, of economical fuel consumption and note the answer. His boilers are running and supplying steam, is not that enough?

Meet the achiever, he is a good man to know. If you really want to "get there," just follow him.

He is a man of character, of grit, has a firm determination to win, and all minus the "swelled head" and the ever-beaten state of exaggeration. His plant shows it. He is "next" to real doings, actual operation; he knows his business from firing to delivering to the board, and he does not have to tell it, either.

To achieve is to win, and to win we must ever play honest. It is the honest game in life that counts.

Let your work be a real achievement. It is bound to pay dividends.

Southern California Edison System

By A. R. Maujer

Historical and descriptive sketch of an electric generating system which eventually will consist of some 12 *hydroelectric plants ranging in capacity from* 1000 *to* 20,000 *kilowatts and two steam-generating plants of which one has a capacity of* 11,500 *kilowatts and the other will have a probable ultimate capacity of* 120,000 *kilowatts.*

The fact that the Southern California Edison Company, of Los Angeles, is building what will be one of the most modern and economical central steam-generating stations on the Pacific Coast should revive the general interest in the extensive system of that company.

The new plant is located at Long Beach where sea water is available for condensing purposes. Long Beach is about 20 miles almost due south of Los Angeles. The ultimate capacity of the plant may be as much as 120,000 kilowatts and the plant will eventually take over the load which is now being carried by the steam-generating station, Los Angeles No. 3, which is located in the city of Los Angeles.

The initial equipment will consist of a 15,000-kilovolt-ampere Curtis vertical turbine and eight Stirling boilers using crude oil as fuel. The steam will be generated at 225 pounds pressure and receive a superheat of 125 degrees. A complete description of this plant will be printed as soon as the construction has advanced far enough for photographing.

A fair idea of the amount of territory covered by the Southern California Edison Company's system may be gained from Fig. 1. The dotted lines to the north on the map indicate future development. The distance between Kern River No. 5, the most northerly of the proposed water-power stations, and Los Angeles is about 140 miles in an air line. The distance between Upper Santa Ana, the most easterly of the water-power stations, and Los Angeles is about 75 miles in an air line. These great distances between the generating stations and the area of distribution are characteristic of many of the water-power systems in the far West.

The first hydraulic-generating station of this vast system and the first three-phase "high-tension" system to be used in America was completed in 1893 at a point about 8 miles east of the town of Redlands in San Bernardino county. The station is now known as Mill Creek No. 1. It contained at the time of its completion two 250-kilowatt three-phase generators driven by Pelton waterwheels at 600 revolutions per minute. The effective head was 295 feet. Current was generated at 2500 volts and transmitted to Redlands at the same pressure. Dr. Louis Bell, who designed the plant, decided upon 50 cycles as the proper current frequency, following the lead established by European practice, 25 cycles for power circuits and the full 50 for lighting.

It is interesting to note that as a result of Doctor Bell's decision in 1892 to use 50-cycle current all public-service current in southern California today is 50 cycles. When the first extensions of the original system were made it naturally followed that the dynamos were wound for 50 cycles so as to be consistent with the existing apparatus. Later when the desirability of 60-cycle current became manifest the system had grown to such proportions that a change over was out of the question. Then, when the Pacific Light and Power Corporation entered the field, in order to be able to have an emergency tie-in with the Edison Company's lines, it also adopted 50 cycles.

In 1896 it became necessary to increase the capacity of Mill Creek No. 1 and this was done by increasing the head and adding a dynamo to the equipment. At the same time three Wagner static transformers were installed to raise the transmission voltage to 10,000 volts, which at that time was considered to be the ultimate in the way of high-tension current possibility. The effective water-pressure head was raised from 295 to 650 feet; this necessitated an increase of 3000 linear feet in the length of the pressure pipe, making the total length

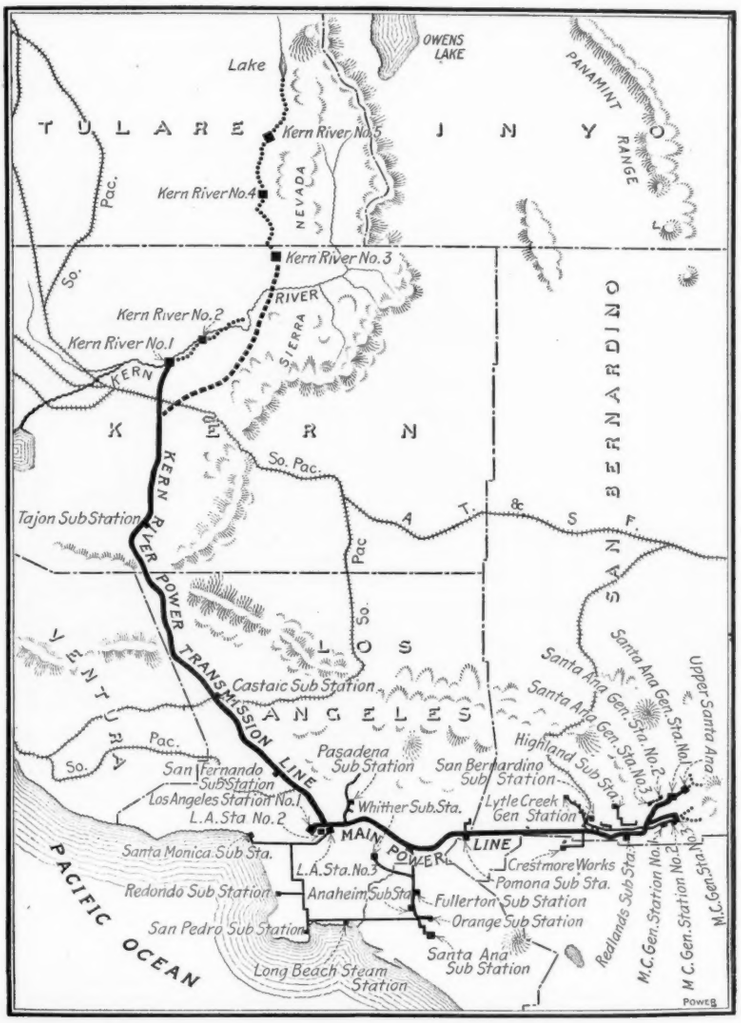


FIG. 1. MAP OF THE SOUTHERN CALIFORNIA EDISON COMPANY SYSTEM

10,250 feet. This increased the capacity of the station to 650 kilowatts. All of the old waterwheels were taken out and new ones, designed to operate economically under the new head, were put in. Water for the wheels which drove the new dynamo was tapped out of the existing pressure pipe.

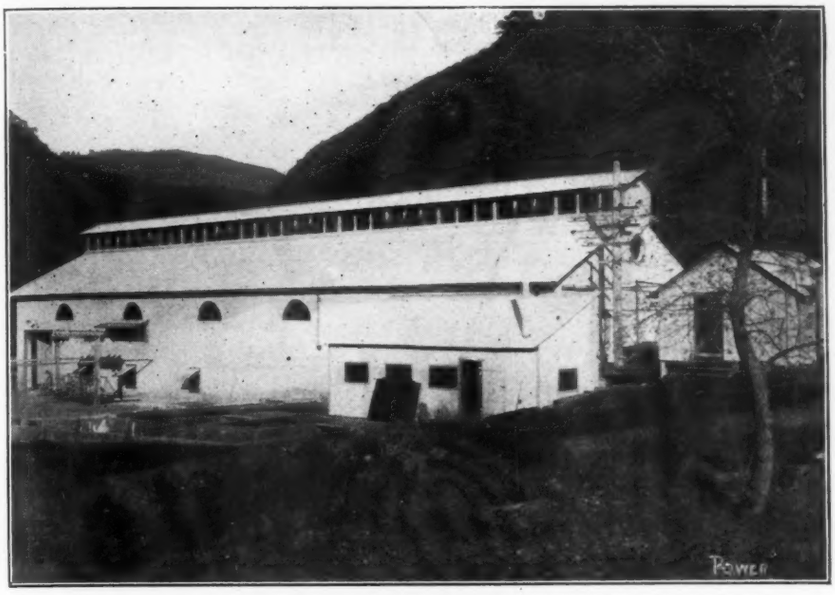


FIG. 2. MILL CREEK NO. 2 WITH ADDITION KNOWN AS MILL CREEK NO. 3

SANTA ANA RIVER DEVELOPMENT

In the latter part of 1898 the Santa Ana River Station No. 1, to the north of the Mill Creek plant, was completed. This station contains four 750-kilowatt General Electric dynamos, of which three are driven by Pelton impulse waterwheels and one is driven by a Doble impulse wheel. The speed is 300 revolutions per minute and current is generated at 750 volts. Originally the plant had but one pressure pipe, but later a second pipe was put in for emergency use. The head is 760 feet and there are 2285 lineal feet of pressure pipe. In addition to this there is considerable wood flume work to keep in repair and some tunnels.

MILL CREEK NO. 2

In 1899 further growth in the demands for power resulted in the erection of Mill Creek Station No. 2, two miles east of the No. 1 station. Here there was installed two 250-kilowatt 11,500-volt three-phase General Electric dynamos driven by Pelton impulse wheels at 375 revolutions per minute. The water-pressure head at this station is 627 feet.

MILL CREEK NO. 3

The next extension to be made was called Mill Creek Station No. 3, although actually it was only an addition to Mill Creek No. 2 and not a separate station at all. Mill Creek No. 3 had the highest head of any hydraulic power plant in America at the time of its erection in 1903. The static head is 1960 feet. Since Mill Creek No. 3 was built one or two plants of still higher head have been put up. The Pike's Peak plant has a head of something over 2000 feet. Mill Creek No. 3 contains four 750-kilowatt dynamos which generate current at 750 volts. Of these, three are driven by Doble wheels and one is driven by a Pelton wheel. The speed is 430 revolutions per minute.

FIG. 3. EQUIPMENT KNOWN AS MILL CREEK NO. 3

At the same time that Mill Creek No. 3 was being installed, the two dynamos in No. 2 were taken out and moved to a new station located a short distance northwest of Colton and known as the Lytle Creek Station. New waterwheels were put on these dynamos as the head at Lytle Creek is only 474 feet while at Mill Creek No. 2 it is 627 feet. All excitation for Mill Creek Nos. 2 and 3 is generated by half of the water supplied to No. 2; the remaining half drives one of the original waterwheels to which has been attached a new 250-kilowatt dynamo.

The pressure pipe for the Mill Creek No. 3 station is 8096 feet long and 26 inches in diameter, reducing to 24 inches at the bottom. Fig. 2 shows the exterior of the combined Mill Creek Stations Nos. 2 and 3 and Fig. 3 shows the four units which are known as Mill Creek No. 3.

SANTA ANA NO. 2

In 1905 the Santa Ana Station No. 2 was completed. It contains two 500-kilowatt General Electric dynamos driven by Doble wheels at 176½ revolutions per minute and generating current at 750 volts. The head is 310 feet and the pressure main is 644 feet long and 36 inches in diameter. There are 7507 feet of tunnel and 2136 feet of siphon pipe. This plant has no flume work.

KERN RIVER NO. 1

The latest and largest of the water-power stations of this system, known as Kern River No. 1, was put into service in May, 1907. This plant is about 100 miles northwest of Los Angeles. It contains four 5000-kilowatt 2300-volt General Electric dynamos driven by Allis-Chalmers impulse wheels at 250 revolutions per minute. The head is 784 feet and there are 1425 feet of pressure main, 6 feet in diameter. To bring the water to this plant, 44,935 feet of canal work are required of which 42,911 feet are tunnels and the balance wooden flumes. The tunnels are 8 feet wide and 7 feet high to the spring of the arch.

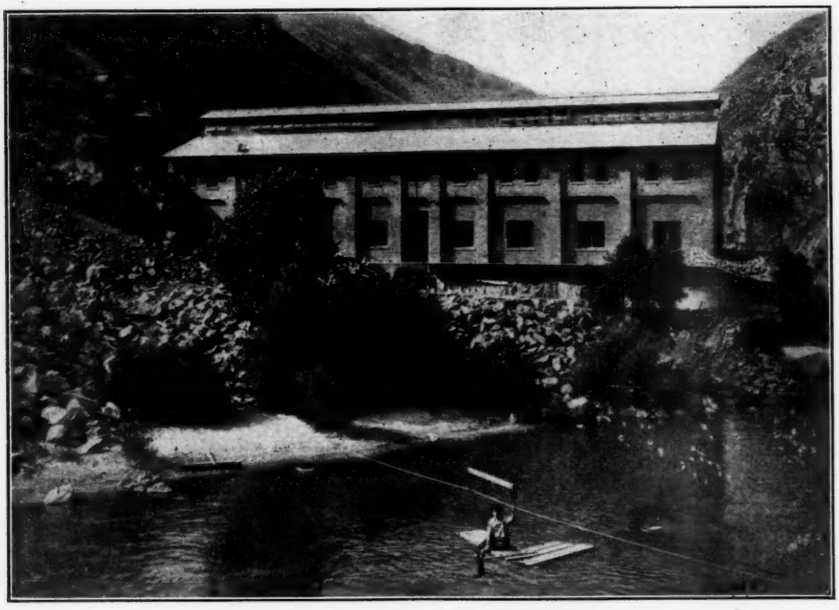


FIG. 4. KERN RIVER STATION NO. 1

STEAM AUXILIARY STATION

As is the case with the big majority of hydroelectric systems, the Southern California Edison Company has a steam auxiliary station. This station, which is designated as Los Angeles No. 3, is located in Los Angeles. Originally it served only as a relay in case of interrupted service from the hydraulic plants, but now on account of the rapid growth in the demands for power part of its equipment generates current even during normal load conditions to piece out the insufficient supply from the water-power plants. When it is ready for service the new Long Beach station will take all of the permanent load now carried by Los Angeles No. 3 and the latter will then be maintained purely as an emergency reserve.

The first two units in Los Angeles No. 3 were put into service in 1904. These are 2000-kilowatt Curtis turbines driving 2300-volt General Electric dynamos at 750 revolutions per minute. The condensers for these units are Wheeler Admiralty type. Originally each condenser contained 6000 square feet of condensing surface but subsequently 3000 square feet were added. The circulating-water pumps are single-stage centrifugal driven by 100-horsepower motors. The condensate is handled by Edwards triplex wet-vacuum pumps driven through steel-rawhide gears by 30-horsepower motors.

Electrical drive was adopted for these auxiliaries because in a relay plant simplicity and reliability of operation are of greater ultimate economy than extreme efficiency with regard to fuel consumption.

In the latter part of 1907 a 7500-kilowatt Westinghouse-Parsons turbo-generator was added to the equipment of Los Angeles No. 3. This unit runs at 750 revolutions per minute and generates current at 16,500 volts.

COOLING TOWER

Perhaps the most interesting thing about the Los Angeles No. 3 station is the cooling tower. This is built in two sections; the first section was put up when the station was built and the second when the No. 3 unit was installed. The tower is of the atmospheric type and built of wood. The first section covers a ground area of 9800 square feet. The filling for both sections is of vertical mats made of common wooden laths. When both of the Curtis turbines are in service about 17.5 second-feet of condensing water pass over the No. 1 section of the tower. The water falls 20 feet.

The No. 2 section covers a ground area of 7490 square feet and contains some 335,000 square feet of wetted surface. In this section the water has a fall of 28 feet. When the Parsons unit

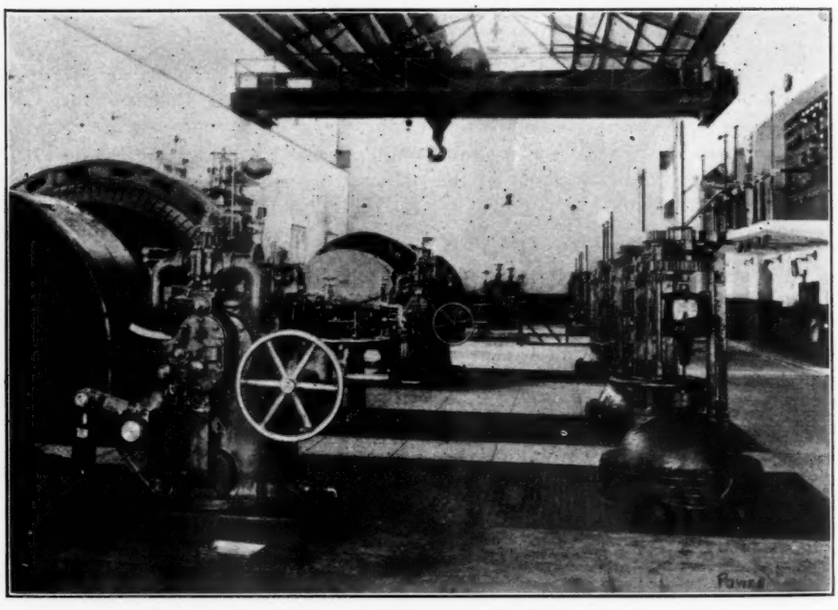


FIG. 5. INTERIOR OF KERN RIVER STATION NO. 1

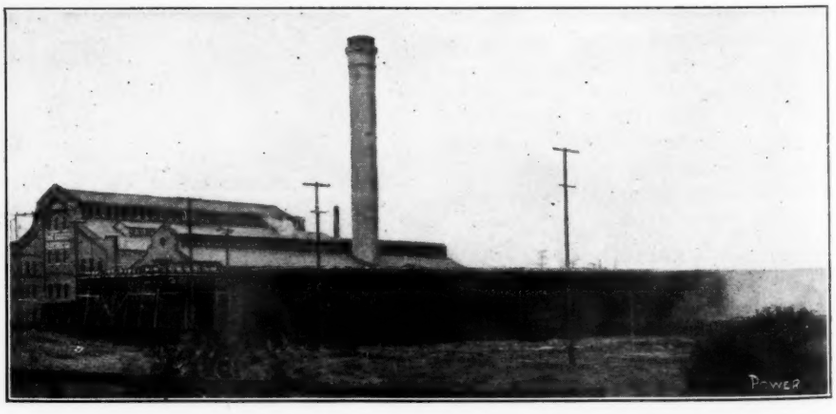


FIG. 6. LOS ANGELES STATION NO. 3, SHOWING PART OF THE COOLING TOWER

FIG. 7. INTERIOR OF GENERATOR HOUSE OF LOS ANGELES STATION NO. 3

is carrying its maximum load 33 second-feet of water pass over the tower.

A recent test of the No. 2 section showed that under maximum-load conditions, with the condenser-discharge water going to the tower at 108 degrees, the temperature of the atmosphere at 65 degrees, and with a humidity of 60 per cent. the circulating water was cooled to 89 degrees and 140,500,000 B.t.u. were dissipated per hour.

BOILERS

The boiler installation for the first two units consists of eight 500-horsepower Stirling boilers, each containing 5020 square feet of heating surface and 702 square feet of superheating surface. Steam is generated at 150 pounds and given a superheat of 125 degrees.

Seven 750-horsepower Stirling boilers were installed to supply steam to the Westinghouse-Parsons unit. These contain 7512 square feet of heating surface and 1600 square feet of superheating surface. They generate steam at 170 pounds pressure and give a superheat of 150 degrees.

Crude oil is used for fuel, fed through Leahy back-shot fuel-oil burners. One stack furnishes the draft for all of the boilers; it is 150 feet high, 12 feet in diameter. The stack is built of reinforced concrete, and is of the plumb-wall double-shell construction.

The feed pumps for the first boilers are Dean duplex, outside end packed, 10 and 6 by 12 inches in size. The pumps for the newer boilers are Snow duplex, outside center packed, 16 and 10 by 12 inches in size.

The boiler makeup water is raised from a well on the property to wooden treating and storage tanks by a 6 and 5¾ by 6-inch Dow duplex pump. The carbonate and sulphate impurities in the water are neutralized with compound manufactured by the company itself.

A 500-horsepower Cookson open feed-water heater serves the original boilers and a 5000-horsepower Cochrane open heater serves the other boilers.

CURRENT DISTRIBUTION

Los Angeles Station No. 3 serves as the receiving station for the current from the water-power plants. The current from the Kern River station is brought into Los Angeles No. 3 at 75,000 volts where it is stepped down for local distribution. The distribution voltages in the city of Los Angeles are 15,000 and 2300 volts.

The transmission voltage from the Mill Creek and Santa Ana groups of stations is 30,000 volts. The current from these stations is sent toward Los Angeles, supplying the various towns along the route. During normal-load periods there is a surplus which is received at Los Angeles No. 3, but during maximum-demand periods there is a deficiency which is made up from the current generated in Los Angeles No. 3. Practically all of the current generated at the Lytle Creek plant is consumed in Colton, 8 miles away. The transmission from Lytle Creek is at 11,000 volts.

We are indebted to R. J. C. Wood, superintendent of generation, for particulars in regard to the equipment of the various stations described herein.

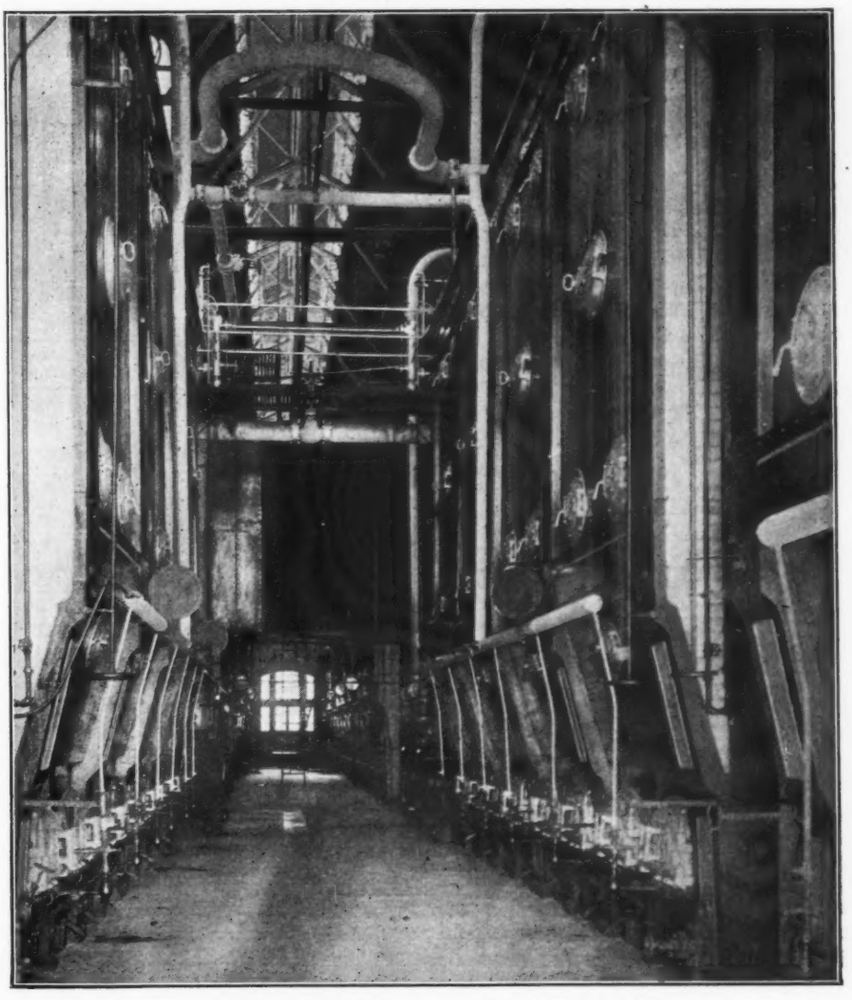


FIG. 8. BOILER ROOM IN LOS ANGELES STATION NO. 3

Increasing Efficiency of Rotary Pumps

By C. A. Tupper

The use of rotating diffusers, invented by Professor Novak of Austria, has been found to increase centrifugal pump efficiency. The idea is also applicable to turbo-blowers and compressors.

During a trip abroad, from which the writer recently returned, he was particularly interested to observe the progress being made in the development of rotary power and pressure machinery, including steam turbines, pumps, compressors and blowers; and in the following will be found the account of an invention affecting all of these. It originated, however, with centrifugal pumps, and has thus far been worked out along that line only, except as noted farther on.

To anyone familiar with such pumps it is well known that their development has been a matter of constant experimenting, in the course of which reasonably accurate rules and formulas have been evolved to cover most of the details of construction; so that now the exact contribution which will be made by any part to the efficiency of the unit as a whole can be very closely determined in advance. Throughout the design the purpose has always been to reduce as far as practicable the frictional resistance of the water and other disturbing influences, thereby converting into pressure as large a proportion as possible of its kinetic energy which arises from the velocity given to it by the impeller.

Among later improvements the most important has unquestionably been the general adoption of diffusers so designed as to form passages leading from the rim of the impeller to the volute, or whirl chamber, the effect of which will be to prevent the formation of eddies and to make the entrance of the water to the volute less of an interruption to its flow.

On the Continent, where the centrifugal pump has developed faster probably than in this country, one of the most eminent designing engineers, Prof. J. Novak, of Prague, Austria, has gone a step farther, if the conclusions drawn in this article are fully substantiated in practice, by constructing a pump with rotating diffusers which he has covered by basic patents.

This pump is comparable with the well known Rateau type, but differs from it in that the diffusers are not fixed but so arranged as to be freely rotated upon the shaft of the impeller or upon a special rigid sleeve.

As a result of this arrangement there are two decided advantages: Reduction of the friction at both sides of the impeller, and provision for an ample whirlpool or diffuser space with least side friction.

The diffuser may consist of one or more shells rotating separately. Its action, or the changing of the kinetic energy into pressure, is accomplished by the lips or sides of the outer edge of the rotating parts. These are so shaped as to form an easy passage from the impeller to the volute. The diffuser shells revolve freely on their axes and the action of the water issuing from the impeller keeps them in motion. Naturally, due to friction of the disks themselves and the decreasing velocity of the water, there is a gradual lessening of the peripheral velocity of these revolving parts as their distance from the impeller is increased. In this way the velocity of the water is gradually stepped down. Bearing this in mind, it is easy to see that the ratio of velocity existing between the water and the revolving diffusers is much less than that produced by the stationary diffusion vanes ordinarily used, and with this re-

TABLE 1. RESULTS OF TESTS OF SMALL VOLUTE PUMP

Test number	1	3	4	5	6	7	8	9	10
Current consumption, kilowatt	1.05	2.0	2.07	2.34	5.1	5.47	2.44	4.39	4.93
Revolutions per minute of motor	1042	1000	1000	1030	990	950	1040	980	980
Efficiency of motor, per cent	...	33	35	39	60	61	39	59.5	60
Delivery of motor, horsepower	..	0.92	0.985	1.24	4.16	4.55	1.293	3.55	4.02
Power consumed by belt, estimated horsepower	..	0.3	0.3	0.3	0.3	0.3	0.3	0.3	0.3
Power consumed by pump, horsepower	..	0.62	0.68	0.94	3.86	4.25	0.995	3.25	3.72
Revolutions per minute of pump	0	2220	2220	2260	2200	2000	2270	2160	2170
Manometric head, meters	..	0	0	15.4	16.6	15.1	17.0	18.8	17.4
Capacity in liters (weir measurement)	..	..	..	0	8.15	9.65	0	8.59	10.32
Delivery of pump, horsepower	..	0	0	0	1.805	1.945	0	2.15	2.40
Efficiency of pump, per cent	..	..	..	..	46.8	45.8	..	66.2	64.5
	..	R	F	F	F	F	R	R	R

NOTE.—Rotating diffuser = R. Stationary diffuser = F. H.P. = Metric horsepower, or slightly less than the United States horsepower standard.

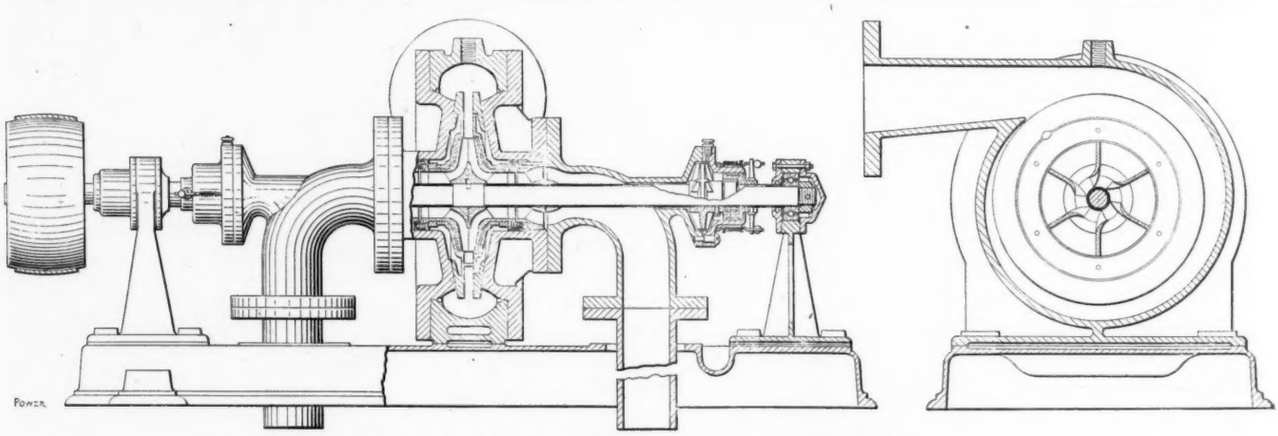


FIG. 1. VOLUTE PUMP FITTED WITH ROTATING DIFFUSER

duction in the relative velocity there is, of course, a very appreciable lowering of the frictional losses.

Table 1 shows the results obtained with the small volute pump represented in Fig. 1, which was the first to be built and which was intended only for experimental purposes. This pump had a 3½-inch discnarge, with two balanced suctions of about 2⅜ inches diameter each, the impeller diameter being about 5¾ inches. The unit was designed to deliver 9 liters of water per second, or the equivalent of about 140 gallons per minute.

It will be noted from Fig. 1 that the pump had a single rotating diffuser and a suitably designed throat. The shells of the diffuser were held together by side bolts, which also assisted the impeller discharge to drive the diffuser by friction along its inner surfaces.

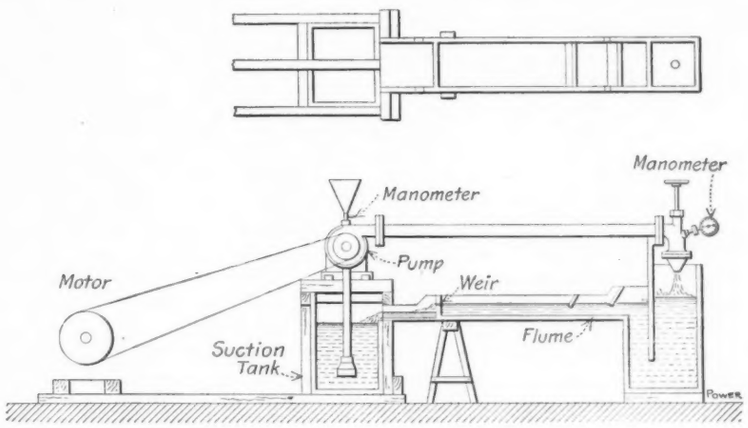


FIG. 2. ARRANGEMENT FOR FIRST TEST

The initial test was conducted by director E. G. Fischinger, a consulting engineer of Dresden, in the interest of the machinery-building and engineering company, Rudolph Meyer Aktien Geselschaft, Mulheim-on-the-Ruhr, which took over the entire patent rights for Germany of the Novak invention.

The arrangement for the test is shown in Fig. 2. A 13-horsepower direct-current motor was belted to the pump and the suction pipe was firmly secured beneath the pump. The water from the suction tank which passed through the pump was led through a tube 8 feet long to an adjustable discharge nozzle. The discharge basin and suction tank were connected by a flume at the lower end of which a Francis overfall measuring weir had been placed. Both the discharge nozzle and the weir were used in determining the volume of water handled, but only the values found with the latter, being somewhat lower than those shown by the discharge nozzle, are entered in the table. For measuring the head there were used two recording manometers, which were compared with each other and the readings of which agreed. In determining the consumption of electric current and arriving at the efficiency of the motor, Director Fischinger used his own instruments, which prior to the test had been carefully calibrated according to the compensation method.

For the tests shown in the columns of Table 1 that are marked *F*, the diffuser was held stationary by an especially provided pin inserted for the purpose. In those marked *R* the diffuser was permitted to rotate freely.

The conditions existing in the operation of a modern centrifugal pump were by no means reproduced merely by holding the diffuser stationary, as the results of the tests themselves show; but the pump was not intended to do more than to indicate the relation between the efficiencies to be obtained from fixed and rotating diffusers. As a specimen of pump construction it left much to be desired, having been built in a plant not adapted to that class of work. Even the rotor was not well balanced, thus rendering its movement very uneven.

Later, another pump built by Rudolph Meyer, Inc., was submitted to tests. These took place in the works of that firm at Mulheim-on-the-Ruhr, under the general direction of Messrs. Kwaysser and Saloman, an engineer in Vienna and an engineer of the Meyer firm, respectively, as well as in the presence of Professor Novak.

This pump had an impeller 14 inches in diameter. The arrangement for the tests is shown in Fig. 3. The results are given in Table 2. From these tests it was evident that better efficiencies could be obtained by lessening the me-

TABLE 2. RESULTS OF SECOND SERIES OF TESTS

Test No.	5	2	1	4	19	7	17	6	3	18	16	12	13	15	9	11	14
Diffuser (fixed or rotating)	ROTATING							FIXED			ROTATING			FIXED			
R.p.m. of pump	1462	1412	1425	1412	1400	1390	1380	1420	1412	1380	1010	1000	1000	1010	1000	1000	1000
Manometric head, meters	44.05	53.4	53.9	52.79	53.29	50.82	41.76	41.11	41.39	34.6	22.02	25.14	23.05	18.82	18.37	19.37	17.83
Capacity per second	0	27.2	27.45	32.1	32.3	37.0	47.2	19.6	23.9	43.2	0	25.6	38.0	0	21.6	22.3	30.2
Consumption of current, kilowatts	10.5	27.8	7.0	29.7	29.0	31.0	37.0	25.5	27.9	36.4	4.99	11.8	13.5	6.00	11.5	11.5	13.0
Motor delivery, kilowatts	8.8	25.2	24.3	26.5	26.0	27.8	33.2	22.8	25.0	32.7	2.83	10.0	11.9	4.75	9.8	9.8	11.3
Pump delivery, kilowatts	0	14.2	14.45	16.5	16.9	18.4	19.3	7.9	9.7	14.6	0	6.32	8.65	0	3.88	4.25	5.3
Efficiency of unit, per cent.	0	51.0	53.5	55.4	58.3	59.3	52.3	31.0	34.8	40.0	0	54.0	64.0	0	33.0	37.0	40.75
Efficiency of pump with belt, per cent	0	56.4	59.5	62.0	65.0	66.0	58.4	34.6	38.9	44.5	0	63.8	73.0	0	39.6	43.5	47.0
Efficiency of pump alone, per cent	0	60.0	62.0	65.5	69.0	70.0	63.4	37.6	42.0	48.0	0	66.2	76.0	0	43.0	47.0	50.0
Hydraulic efficiency, per cent	78.0	82.0	82.0	80.0	84.0	81.0	68.0	62.5	63.0	56.0	79.0	74.0	72.5	67.0	52.0	58.0	52.0

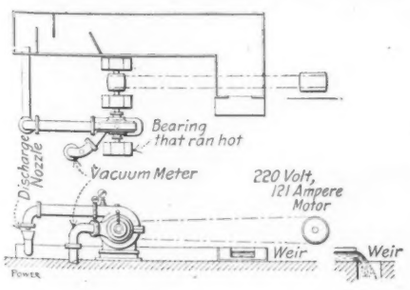


FIG. 3. ARRANGEMENT FOR SECOND TEST

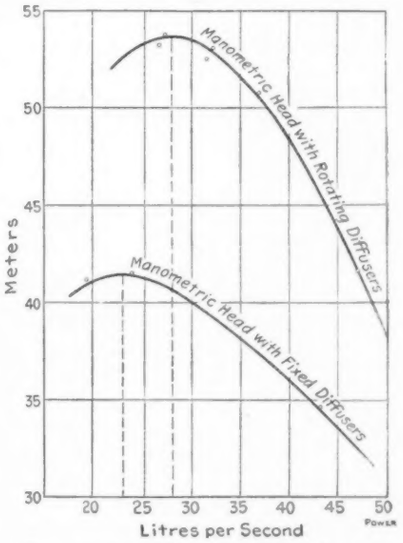


FIG. 4. PUMP PERFORMANCE SHOWN GRAPHICALLY

chanical losses, and two rings were taken from the center bearing which had been continually running hot.

For four hours on the following day a third test was then run. During this the manometer remained at a constant hight, except for some slight fluctuations

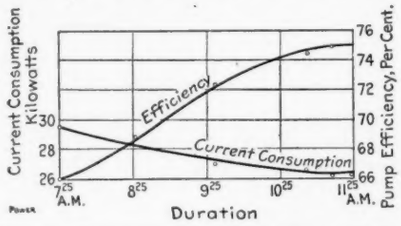


FIG. 5. EFFICIENCY AND CURRENT CONSUMPTION SHOWN GRAPHICALLY

caused by differences in the speed, and a steadily increasing volume of discharge was observed, with a lessening in the consumption of electric current and a rise in the efficiency, which reached 75 per cent. In Figs. 4 and 5 this is shown graphically.

Since then additional pumps have been built and the writer has been informed that they are showing good results in service; but their manufacture has not yet reached the commercial stage as Professor Novak is proceeding slowly in the development of the various sizes and wishes to standardize them before putting the pumps on the market.

In the United States the simultaneous development of the pump has been undertaken by O. C. Goeriz & Co., of San Francisco, Cal., who are just completing their first unit, a single-stage house pump, with double inlet, for 100 gallons per minute delivery against 55 pounds pressure, or the equivalent of 120 feet head. This will run at 3400 revolutions per minute and is to be operated by a direct-connected 5-horsepower alternating-current (two-pole) motor. Drawings for similar pumps of 50 and 150 gallons per minute are now in course of preparation.

This firm makes interesting comparisons between a twin-centrifugal pump of standard design, shown in Fig. 6, and a similar pump redesigned in accordance with the Novak system, as illustrated in Fig. 7. The company says:

"It is known that the frictional resistance between a revolving disk and water depends on the squared relative velocity, while the energy loss must conform with the cubed ratio of the relative velocity between the disk and water. Applying this to a comparison of disk friction for the pump shown in Fig. 6 as against that in Fig. 7, and assuming that the runner in both cases revolves with the same speed, the following conclusion can be reached: In the former the water volumes *B* and *C* can be assumed to run at half speed, which is the middle between the full-speed runner and the stationary side walls. The relative velocity, or difference between full and half speeds, is therefore one-half speed. In the case of Fig. 7 the diffuser *DE* revolves at half speed itself, as proved by the tests made with Novak pumps; then the water volumes *B* and *C* are churned with, say, three-quarter speed, or the relative velocity is one-quarter.

"For equal impeller diameters and equal impeller speeds it would therefore follow that, because the relative velocity between the impeller sides and the water in Fig. 7 is one-half that in Fig. 6, the frictional resistance must be one-quarter, and the loss of work by friction must be only one-eighth of what losses occur in the ordinary case represented by Fig. 6. The energy losses in chambers *F* and *G* do not sum up with the losses in *B* and *C*. What waste is going on in *F* and *G* does not directly affect the energy required at the pump shaft. However, it influences the velocity of the diffuser halves *D* and *E* and thus indirectly the waste going on in *B* and *C*. The friction loss in *B* drives the diffuser

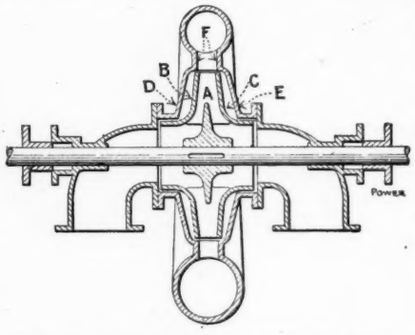


FIG. 6. STANDARD TWIN-SUCTION CENTRIFUGAL PUMP

and the loss in *E* acts like a brake, but both have to balance; otherwise the diffuser must gain or lose speed until such equilibrium of friction forces shall be attained.

"This is one part of the Novak improvement. Contemplating the other and more important innovation, the two sketches, Figs. 6 and 7, must be compared again.

"In Fig. 6 the water rushes from the impeller at a velocity which, for the sake of convenience, may be termed 'full speed,' and along the walls *F* of the stationary diffuser a certain part of the energy contained in the water must be dissipated by disk friction. The relative velocity between water and side walls is the entire full speed. Now, in Fig. 7 the walls *K*, which belong to the diffuser halves *D*, and *E*, revolve again at half speed, and the relative velocity can be expressed as half speed. Therefore the frictional resistance in the arrangement shown by Fig. 7, as against that of Fig. 6, assuming equal extension of the diffuser walls in radial direction and similar conditions of the surface, will be one-quarter, making the loss of energy one-eighth only."

The invention of Professor Novak is adaptable either to a volute or a turbine pump, and the selection of type depends mainly upon the head against which the water is to be delivered.

The considerable reduction of side friction and the improved whirling action in the Novak pump make it possible in many cases to run the impeller at a higher rate of speed, and thus as compared with a multi-stage unit to perform the required service with a single impeller without sacrifice of efficiency. For example, a one-impeller pump with the Novak diffuser can be made to replace a four-stage turbine pump of standard design running at half speed. Such a single impeller, which may require a casing of considerable radial extension if surrounded by several Novak diffuser shells, connected with a motor running at higher speed, will therefore be a strong competitor of the low-speed (or even equal speed) multi-stage turbine pump. Thus it will be practicable to build pump units, including higher-speed steam turbines designed for the most favorable conditions of steam consumption, with corresponding benefit to the net efficiency in the latter direction also.

The test results shown in the tables indicate furthermore that an excellent standard line of house pumps can be developed at relatively low costs to give high efficiency, thus putting them in the lead of triplex pumps, as the latter require a number of gears between the crank shaft and an electric motor operating at lower speed.

For high-lift pumping, as in mines, fire-protection systems, etc., the 100 to 300 pounds pressure attainable with pumps built on the Novak design will

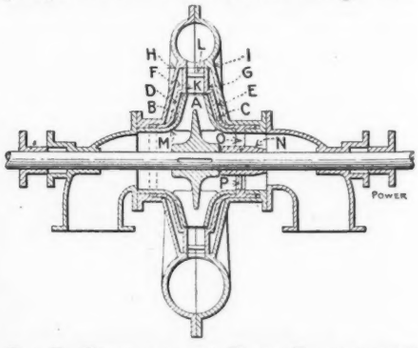


FIG. 7. TWIN-SUCTION PUMP FITTED WITH ROTATING DIFFUSERS

offer additional opportunities for its application.

Furthermore, it is apparent from Fig. 7 that the Novak type of pump can be conveniently standardized, inasmuch as even a considerable difference in service requirements can be taken care of by a slight alteration to the first revolving diffuser, thus adapting the same size to smaller or larger lifts.

Certain mechanical improvements which have been brought out by the American representatives of Professor Novak, but which they state fall under the domain of his basic patents, are indicated on the right side of Fig. 7. Here a bushing *N* is shown, which is loose on the shaft but rigidly connected to the Novak diffuser half *E*, either by vanes *O* similar in shape to propeller blades or merely by stays *P*, thereby supporting the whole diffuser directly by the shaft itself. By this alteration a material reduction of mechanical friction is gained. Furthermore, if the water handled is gritty, oil or grease can be readily fed through a bore in the shaft directly into the clearance between bushing *N* and the shaft. In addition, provision has been made, it is stated, to so construct the impeller and diffuser that end thrust is taken care of automatically.

"It is the logical inference," says Director Fischinger, who made the tests first above mentioned, "that moving diffusers will have an influence upon the efficiency of rotary blowers and compressors equally as great as that which has here been established in relation to centrifugal pumps. Should that inference be confirmed by practical tests, then there will be a great future opened up for the turbo-compressor."

Professor Novak is now working on the application of his patent to air compressors of the Rateau type, and some interesting developments in this line are expected to be announced shortly. Rud. Meyer Aktien Gesellschaft have already built and exhibited Rateau turbo-compressors fitted with Novak diffusers and are putting them on the market.

Water Power Dam at Keokuk

By S. Kirlin

The dam is to be 32 feet high and nearly a mile long. It will be surmounted by adjustable steel flood gates 11 feet high and the available head will vary from 31 to 39 feet. The initial installation of waterwheels will have a capacity of 120,000 kilowatts, though the ultimate capacity will be 220,000 kilowatts.

A large water-power project that has attracted a great deal of attention throughout the Middle West for some time is that of the Mississippi River Power Company, which includes a dam across the Mississippi river between Keokuk, Ia., and Hamilton, Ill. The work has now reached the stage where a good idea can be had of the magnitude of the undertaking.

The dam is being built at the foot of the Des Moines rapids, which have always constituted an obstruction to navigation at this point except at times of high water and around which the Government has maintained a system of locks which was constructed in 1877. The new dam will make the maintenance of these locks unnecessary, as the power company will build new locks of much greater capacity, and these will be furnished to the Government free of cost. The new locks will be designed for a channel depth of 8 feet if the Government decides on that depth in the river channel. The dam will also furnish a permanent navigable stage of water for a distance of 60 miles above.

The dam, including abutments, will have a total length of nearly one mile, the spillway section being 4278 feet in length. Its hight will be 32 feet above the river bed and the width of the base 42 feet. The upstream face will be vertical; the downstream side will be curved, having an arc of a circle at the toe which will throw the water away in a horizontal direction. On top of the spillway will be placed 119 steel flood gates 30 feet wide and 11 feet high, supported by concrete piers 6 feet thick, built integral with the dam. They will also support an arched bridge from which the gates will be electrically operated and so manipulated that the water above the dam will be maintained at practically a constant level at all seasons of the year.

The dam will be built entirely of concrete, without any reinforcement, and will be locked firmly in the river bed which is formed entirely of solid blue limestone at this point, making an ideal foundation for this type of structure. All work on the dam proper is being done on the Illinois side of the river, while the power house and locks are being built from the Iowa side. Two temporary power houses, one on each side of the river, furnish the power for operating all engines, pumps and other

FIG. 1. TWO SECTIONS OF THE POWER-HOUSE COFFERDAM

machinery used on the works. The one on the Iowa side contains 1000 horsepower in boilers, one 350-horsepower air compressor and one smaller compressor having a capacity of 150 horsepower. An electric-lighting plant is located in this power house in order that the construction can go forward night and day.

On the Illinois side the temporary power house contains 800 horsepower in boilers, these furnishing the steam for operating all engines, pumps and drills used on the Illinois side of the river. An unusual feature of the work is the use of compressed air in all of the engines driving the different apparatus, the engineers deciding that this would be the most economical considering the scattered position of a large portion of the machinery to be driven.

The capacity of the concrete plant is 1200 cubic yards per day and involves the moving of 100 carloads of rock, 50 cars of sand, 6 cars of cement and 100 trains of concrete per day. About 300,000 cubic yards of sand will be required to complete the work, which is obtained by means of a large sand-pumping plant located in the old channel of the Des Moines river about two miles below the dam. Great care is exercised that all material is up to standard grade, a complete laboratory being maintained on the Illinois side for this purpose. Samples of the concrete mixture are taken at frequent intervals from the buckets which convey the material to the forms. All cement is carefully tested by the company's representative at the mills before it is shipped to the warehouses at the works. All stone used in the concrete is blasted out of the river bed in making the foundations, and is of the finest quality.

The power house and locks will be located entirely in the bed of the river on the Iowa side. The power house will be 1400 feet long and 123 feet wide, and it will extend at right angles to the dam. The substructure will be of massive concrete in which will be molded the water-wheel chambers and passages leading to them. On top of this will be built the superstructure, of concrete, brick and steel construction; it will contain the generators, transformers and switchboards. There will be 30 vertical generating units; the generators will be mounted on the upper end of the shaft and each shaft provided with two turbine wheels at the lower end. The hight of the power house from the foundation to the roof will be 133 feet. The quantity of stone which it is necessary to blast out of the bed of the river in locating the foundations for the power house and lock is almost exactly the amount needed in the concrete work of the entire structure, including the dam, power house and lock. This rock, after it is blasted out, is loaded into cars by means of steam shovels, transported to the crushing plant and when crushed is conveyed to the bins over the mixing plants located on each side of the river.

FIG. 2. LOOKING TOWARD IOWA SHORE

To prevent floating ice, logs and other objects from entering the channels leading to the wheels, a concrete wall will be built from the upper end of the power house to a distance of 2800 feet upstream, at which point it will curve in to a junction with the shore. This wall will contain a large number of openings

FIG. 3. LOOKING TOWARD ILLINOIS SHORE

beneath the water level to allow the water to flow freely to the forebay and thence to the wheels, but it will shunt all ice, logs, etc., around through the spillway, and thus practically eliminate all interruptions to the service.

The method of construction involves the use of the dam itself as a base for the tracks employed for transporting material. Three tracks are built out from the abutment on the Illinois side. As these tracks can be extended only as far as the concrete has hardened it was necessary to use some other means for conveying the material from this point to where the new concrete was to be deposited in the forms. This is done by means of the two large dam travelers, shown in the illustrations. These are massive steel cranes provided with three trolley tracks extending 210 feet out over the dam from the point where they rest upon the hardened concrete. Each of these runways is equipped with a trolley, tackle and carriers operated at high speed by two reversible engines located at the rear end. These engines are operated with compressed air and are under the control of one man.

The travelers are mounted upon six heavy cast-steel track wheels running upon a specially designed 100-pound rail track laid on I-beams firmly embedded in the concrete forming the dam. The concrete to be deposited in the forms is run out onto the dam by trains carrying a large number of buckets, each holding 1½ cubic yards. At the rear end of the travelers these buckets are picked up by the trolley tackle and rapidly conveyed to the end of the track which overhangs the steel forms to be filled. These buckets, which can be dumped at any point of travel, are conveyed by endless lines running over spool-drum rigs, each being operated by a separate reversible engine. The hoisting and lowering operations are accomplished by means of single-running block rigs operated by a more powerful engine. As the work progresses the rails are extended on the hardened concrete and the travelers moved forward. The forward posts of the travelers are 65 feet high, and the entire weight of each machine is 175 tons, including all necessary equipment.

A large amount of cofferdam work is necessary to unwater the river bed on which the foundations are being built. On the Iowa side, where the foundations are being installed for the power house and locks, it was necessary to unwater 33 acres of river bed. The cofferdams are composed of a large number of timber cribs 24 feet long and 16 feet wide. They are filled with stone and sunk to the river bed at intervals of 12 feet around the entire space that is to be unwatered. When the cribs have been placed the space between each is closed by lowering square timber stoplogs, after which the entire outside surface is sheathed with a layer of planks against which earth is deposited, the earth filling the small cracks between the planks and making the cofferdam water tight. A layer of riprap over the earth prevents it from being washed away. After the cofferdam is completed the space inclosed is pumped out by large centrifugal pumps which throw out the water at the rate of 10,000 gallons per minute. The average hight of the water outside the cofferdam is about 10 feet.

A vast amount of cofferdam work is necessary on the project, and there has probably never been a more efficient one built than that which surrounds the space on which the foundations for the power house and lock are being constructed.

In building the foundation for the main dam the river bed along the line of the dam will be unwatered in sections by cofferdams 400 feet long and wide enough to allow blasting out the foundation. After completing a section of the dam, another section of the same length will be built ahead, pumped out, and completed in the same manner. As the different sections are completed the flood-gate openings will allow the water to flow freely through the completed portion of the dam. After the entire dam is finished these flood gates will be closed, and the water will rise to the full hight of the dam.

No part of the work is being done by contract, the men being employed directly by the chief engineer. All of the machinery used in the construction work was designed by the engineering department of the company under the chief engineer's supervision.

It is estimated that the initial installation of 120,000 electrical horsepower will be available for distribution by July, 1913. Of this amount, 60,000 horsepower has been contracted for by the St. Louis Public Utilities under a 99-year contract; the remaining 60,000 horsepower is to be sold in the cities near the plant. The head of water will vary from 31 to 39 feet at the different stages of river flow. The maximum amount of power that can be developed is estimated at 220,000 horsepower, and the company will install additional generating units as the demand for power increases in the surrounding cities and towns.

Two New Types of Dynamometer*

By Dr. Alfred Amsler

One dynamometer utilizes a flexible shaft the torque of which serves as the index to the power transmitted. The other uses an arrangement of tight and loose pulleys; power is transmitted through pistons on the loose pulley acting on oil in cylinders on the tight pulley.

*Abstract of paper read before The Institution of Mechanical Engineers at Zürich, July, 1911.

It is the author's intention to confine himself to describing two types of transmission dynamometers constructed by his own firm. One type, the torsion dynamometer, is designed for measuring the power transmitted to or from high-speed machines, such as centrifugal pumps, fans, turbo-compressors, steam turbines, dynamos, etc., where the torque is fairly constant and therefore need not be recorded in a diagram. The other type, the hydraulic dynamometer, is intended to be used for measuring and recording the energy absorbed by slow-running machines of variable resistance, such as machine tools, plunger pumps and the like.

Torsion Dynamometer

Fig. 1 illustrates the torsion dynamometer which may be used to couple the shaft of the driving engine directly to the driven machine. If desired, pulleys may be fitted to the ends of the dynamometer and the power may be transmitted to one end by means of a belt and taken off at the other end also by a belt. The ends of the driving and driven shafts are securely coupled together by the shaft *G*. This shaft has therefore to transmit the entire power from the one machine to the other, and consequently it is subjected to torsion and is twisted. The shaft is, of course, so designed that the yield point is not exceeded. Thus the shaft works as a torsion spring. Provided the shaft be perfectly elastic, the angle through which the opposite ends are turned will be proportional to the twisting moment; that is, proportional to the power transmitted. If the shaft is made of a special spring steel of very high yield point; that is, above 90,000 pounds per square inch, it can be twisted to a considerable degree without breakage.

The various parts attached to the shaft *G* serve partly for indicating its angle of twist and partly to prevent it from being twisted excessively by being overloaded. For reading the angle of twist,

three disks *M*, *N* and *O* are used. The disk *M* is fixed firmly to the end *H* of the shaft; *N* and *O* are fixed to the end *F* of the shaft. The disk *O* is provided with a radial slit *P*. To the disk *M* a transparent rim *U* made of celluloid is fixed on which divisions are cut. Opposite to the slit *P* a small window has been cut out in the disk *N* and is provided with a fine slit *T*.

When the eye at *Q* looks through the slit *P* the slit *T* will be seen as a streak of light and the divisions will show black on the scale *U*. The slit *T* serves as a pointer for indicating the relative motion of the two disks *N* and *O* as compared to the disk *M*. The line of vision is perfectly defined by the two slits *P* and *T*; parallax is therefore impossible when reading the scale and the observation is therefore independent of the distance between the scale *U* and the slit *T*.

When the apparatus is stationary it will be clear that, if the shaft is twisted, the angle of twist will be shown by the movement of the pointer *T* over the divisions on *U*. But this pointer and divisions will also be visible when the instrument is running; in fact, they will be clearer and more defined than when the instrument is at rest, for while it is necessary to place the eye close to the slit *P* when the instrument is at rest to see the pointer and the scale, it is possible when the instrument is in motion to read the scale at some distance from the slit *P*. The greater this distance is, the more the scale will appear to be enlarged; this enlargement, it should be noted, is independent of the speed. The reason why the scale is still seen when the machine is running is due to the following: The eye looking at the scale through the slit *P* sees the pointer and scale once for every revolution, and only during the very short time when the small slit *P* is passing before the eye; the light impression is therefore instantaneous. One of these impressions would be too small to leave a permanent image in the eye, but since a number of impressions is given to the eye in rapid succession, namely, once during each revolution of the disk, the eye appears to receive a continuous stationary image of the pointer. Should the machine work at a speed below 250 revolutions per minute these images are not continuous, and the reading of the divisions becomes cumbersome, although still possible.

To obtain a permanent impression on the eye, the scale must be subjected to a strong light. The short period of observation must be strengthened by the intensity of the light, since a fixed minimum amount of light energy, that is, intensity multiplied by duration, is required to affect the optic nerve. A 50-candlepower lamp with a ground-glass globe suffices for illuminating the scale and should be placed as close as possible to and behind the disk *M*. Instead of looking at the slit *P* directly it will be found better to insert a mirror so that the scale can be observed from the side as shown in Fig. 1. That the eye be not dazzled, the daylight should be screened off. The observations can best be made in a dark room and only one eye should be used, since each eye would tend to form a separate image which would blur the actual reading. If the head is held so that the line connecting the two eyes is parallel to the slit *P*, the two separate images will coincide. The angle of torque thus observed shows the twist strain in the shaft, since the slit *P* passes the eye at regular intervals. Should the power transmitted and consequently the twist of the shaft vary rapidly, the eye will see different parts of the scale in rapid succession and the image will become blurred, making it impossible to read the scale accurately. When high-speed machines are used, this disturbance need not be feared, while the inertia of the machine itself and that of the disks *M*, *N* and *O* of the dynamometer will restrain any oscillations thus formed. If the twist varies periodically in synchronism with the rotation of the shaft, the wave of twist variation being repeated, every revolution can be followed by the eye by looking at the scale through the slit at different points around the dynamometer shaft.

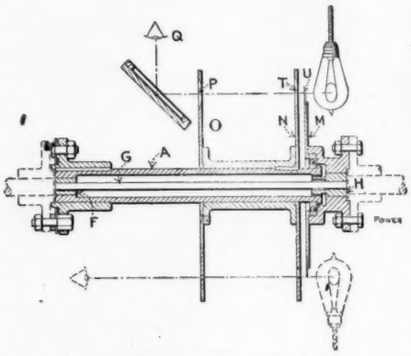


FIG. 1. TORSION DYNAMOMETER

Since the shaft and the other parts connected to its extremities have no tendency to move due to the centrifugal force, the readings of the dynamometer are independent of the speed. The instrument can therefore be run at any desired speed, provided no forces are set up greater than can be resisted by the material of which the disks are made.

The instrument is made generally for speeds of about 4500 revolutions per minute, but some instruments have already been made for 7000 and even 8500 revolutions per minute. The coefficient of elasticity of the shaft can be determined when at rest by fixing a lever of known length at one end and loading this with weights.

The largest instrument at present made can measure torques up to 130,200 inch-pounds and the smallest up to 694 inch-pounds. The instrument suffers from but one defect, namely, the air resistance of the rotating parts. For the time being, it need merely be stated that the influence of the air is very small, especially when the dynamometer is so inserted that the disk *M* is placed nearest

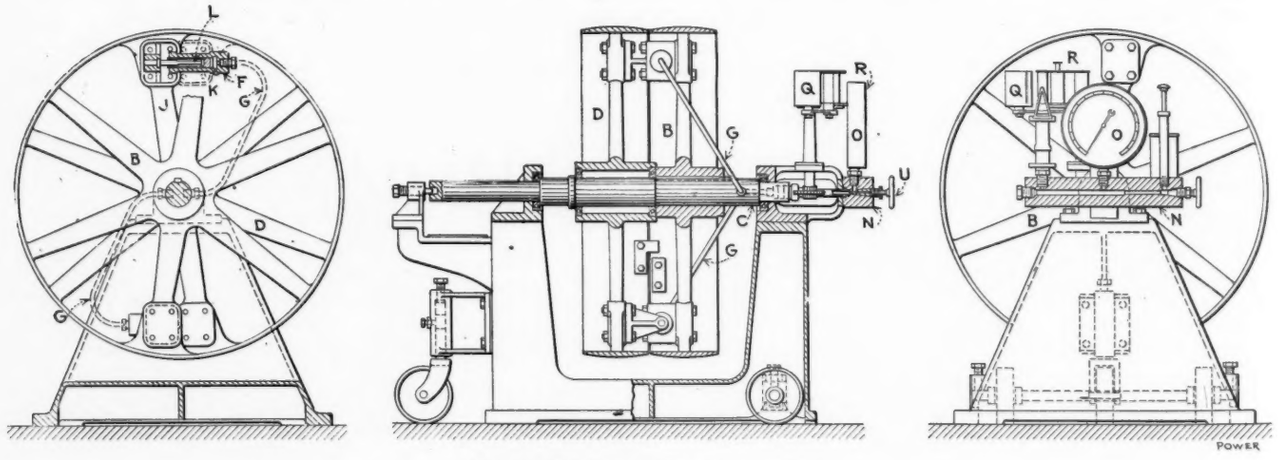


FIG. 2. HYDRAULIC DYNAMOMETER

to the machine the power of which is to be measured. It is possible, although somewhat laborious, to make accurate determinations of the influence of the air resistance on these readings. The dynamometers are so constructed that they can without any difficulty be placed or taken out from between the driving and driven machines, and another shaft can readily replace the former without having to move either of the machines.

Hydraulic Dynamometers

The hydraulic dynamometer, which is used for measuring and recording the power absorbed by machines subjected to a variable load, is illustrated in Fig. 2. The dynamometer is driven from the engine by a belt and transmits the power by a second belt to the driven machine. The driving and driven pulleys *D* and *B* are placed close together on a common shaft. The pulley *B* is fixed to the shaft *C*, while the pulley *D* runs loose on the shaft without any appreciable friction. To the pulley *B* the two cylinders *F* are fixed, which are connected by means of the two tubes *G* to the center of the hollow shaft *C*. The pulley *D* is provided with two projections *J* which press against the piston *L*. The piston works without any friction in the cylinder *F* owing to the omission of any packing, and the fact that the entire system is filled with oil. When the pulley rotates in the proper direction, the projection *J* presses back the piston in the cylinder *F*, thus exerting a pressure on the oil. This oil pressure is transmitted through the hollow shaft *C* to a fixed tube, the end of which fits axially into the shaft *C* and is properly packed by means of a stuffing box. The pressure is further transmitted to the casing *N*, to which on one side a pressure gage *O* is fixed and to another side the recording apparatus. The pressure gage indicates the torque with which the pulley *B* drives the pulley *D*, since the latter is proportional to the oil pressure. The transmission of the oil pressure to the recording apparatus is quite similar to that of a steam-engine indicator; the paper strip is rolled up in the hollow drum *Q* and passes under the pencil of the recorder and is then rolled up on the drum *R*. The paper strip passes under the pencil at a velocity proportional to the speed of the shaft *C*. In order that the number of revolutions of the shaft *C* may be checked readily, a small hammer is attached which gives an audible tap at every tenth revolution of the shaft *C*. These taps can readily be timed by means of a watch. The area of any portion of the diagram represents the energy absorbed by the machine, and the mean hight, that is, the area divided by the length of the diagram, the average power transmitted.

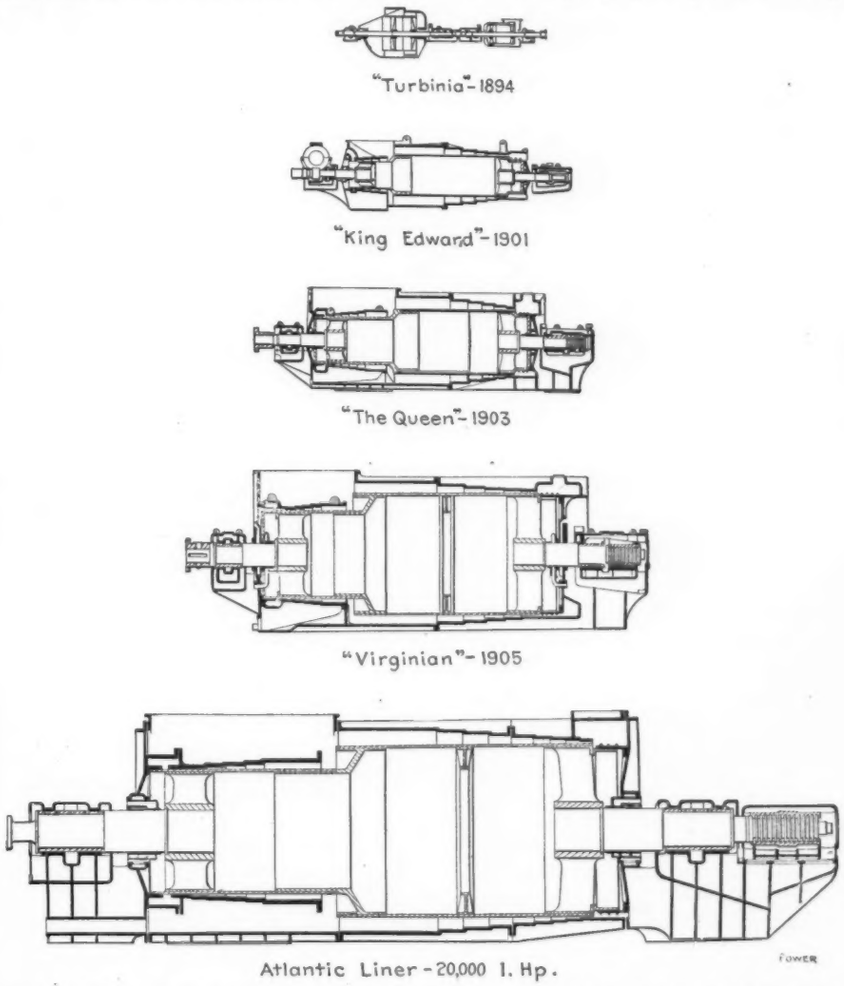
If the resistance of the machine, the power absorption of which is to be measured, is very irregular, the pointer of the pressure gage will oscillate violently and the recorder will draw a zigzag line. These oscillations can be readily reduced by closing the valve *U* to a greater or lesser degree; by so doing the opening through which the oil enters through the shaft *C* into the casing *N* can be increased or decreased as desired. This dynamometer is not nearly as simple and does not indicate the power as directly as the torsion dynamometer; it is consequently less accurate and needs watching. On the other hand, measurements can be made which are impossible to obtain with the torsion dynamometer, namely, the power is continuously recorded on a strip of paper and can be accurately determined, however much it may vary.

The readings given with the hydraulic dynamometer are not entirely independent of the speed; the centrifugal force tends to drive the oil in the tubes outward and to move the piston *L* axially, according to the position in which the pistons are relative to their respective cylinders. By means of a suitable arrangement of the pistons and cylinders with regard to the pulley *B* it is possible to compensate these two forces; this is only possible, however, for one position of the pistons, namely, the middle position. If the dynamometer is allowed to run at a low speed, the disturbing action of the centrifugal force in the pistons when no complete compensation is obtained becomes negligibly small. Another reason for keeping the speed of the instrument low is on account of the centrifugal force causing the pistons to rub against the sides of the cylinder. The dynamometer illustrated may be used for torques up to 13,020 inch-pounds.

The same principle of construction can be applied for measuring higher powers; the instrument is then simply placed in series with the transmission shaft, and instead of the pulleys *B* and *D*, couplings are used. It is, however, necessary in this case that one shaft should be hollow for its entire length so that the hydraulic pressure can be measured at the end.

Growth of Marine Turbine

The accompanying diagram, showing the growth of the marine turbine from its first successful use in the "Turbinia" in 1894 to the present, is reproduced from a reprint in *Engineering*.



The paper was read before the Institution of Naval Architects by the Hon. Charles A. Parsons, of which body the author is vice-president.

Electrical Department

Especially conducted to be of interest and service to the men in charge of the electrical equipment

Drying Out a Flooded Substation

By A. D. Blake

A submerged motor is not an uncommon thing in power-plant practice, but the spectacle of a substation entirely under water is a very unusual one.

The substation herein discussed is located in a sump (see Fig. 1) some 50 feet below tide-water and handles the drainage from a deep cut covering an area of two square blocks; the water is pumped up into the sewer through a 24-inch discharge pipe against a head of 40 pounds pressure per square inch. The equipment consists of three 11,000 to 550-volt oil-cooled transformers of 75 kilovolt-amperes each, two 100-horsepower three-phase vertical induction motors directly connected to centrifugal pumps, a high-tension switch structure containing duplicate sets of disconnecting switches, oil switches, series transformers and relays, and two float-operated starters for the motors.

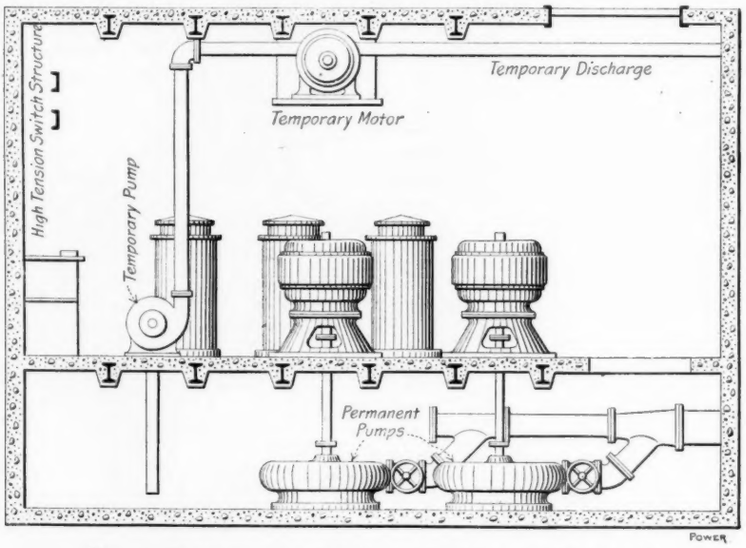


Fig. 1. Elevation of Apparatus before the Flooding

While the permanent apparatus was being installed, the water in the sump was handled temporarily by a 4-inch centrifugal pump belted to a 20-horsepower induction motor and the latter was suspended from the ceiling to leave more clear space on the floor. This temporary pump had kept the water down satisfactorily for a number of weeks, during which period there had been several rains and one thaw. This created a certain feeling of security in the minds of those in charge, and therefore no additional precautions were taken. But, as is often the case, the unexpected happened. There occurred an unusually heavy rain one night, and the water began to come in faster than the pump could take it out. In spite of all the efforts of the operating force, by morning the water had risen to within 2 feet of the ceiling, submerging not only the permanent apparatus but the temporary motor as well.

The conditions were rather discouraging when the writer arrived the next morning, especially as the installation was about 90 per cent. completed and it was now evident that several weeks would be required to undo the damage that it had taken the rain only a few hours to accomplish.

The first thing to be done, of course, was to get the water out. Fortunately, there was an air line close at hand, so an air pump was obtained and after some hours the sump was emptied. Everything was covered with mud and slime, necessitating a washing down with clean water. To do this, the transformers had to be lifted out of their cases and thoroughly cleaned, the cases washed out and dried and the transformers replaced.

The problem now confronted us of the quickest and yet most effective means of drying out, with the limited facilities available. It was finally decided to use both internal and external heat where possible. A medium-size cylinder stove was procured and housings and flues of sheet iron were con-

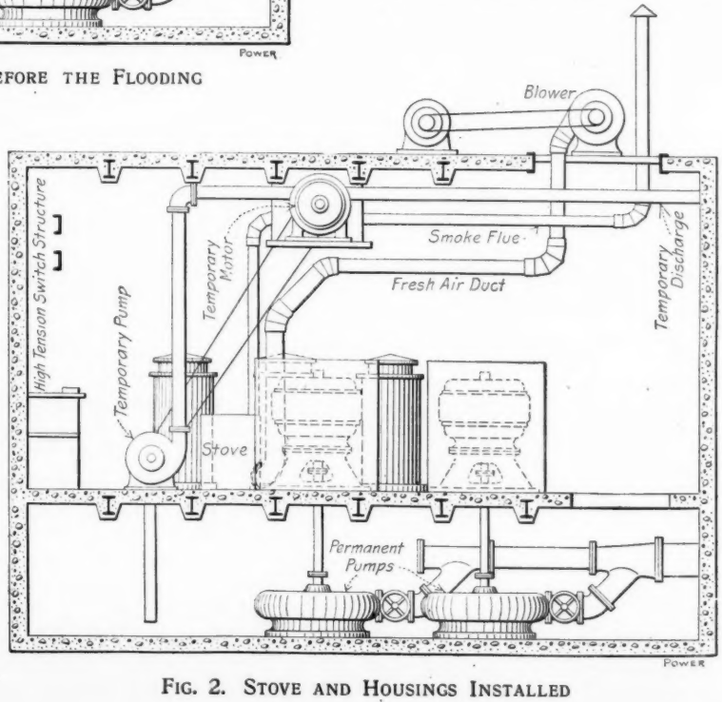


Fig. 2. Stove and Housings Installed

structed as shown in Figs. 2 and 3. A small blower connected to a 2-horsepower motor supplied fresh dry air to the system while the moist air escaped through the hatch at the top of the sump. Thus a positive circulation was established, the air being forced into the transformers through the 2-inch oil drains at the bottom of the cases.

In addition to the use of warmed air, internal heating of the transformers was accomplished electrically, the connections being as shown in the diagram, Fig. 4. The large transformers were connected Y-fashion on the primary side to the 400-volt temporary line, with their secondary windings short-circuited. In series with one of these were connected three of the small series transformers with their secondaries also short-circuited. A voltmeter and an ammeter were connected in the circuit and thermometers were inserted between the transformer windings. Readings were taken every hour and a temperature of slightly under 80 degrees Centigrade was maintained. This required a primary current of 4.2 amperes, which induced 88 amperes in the secondary windings.

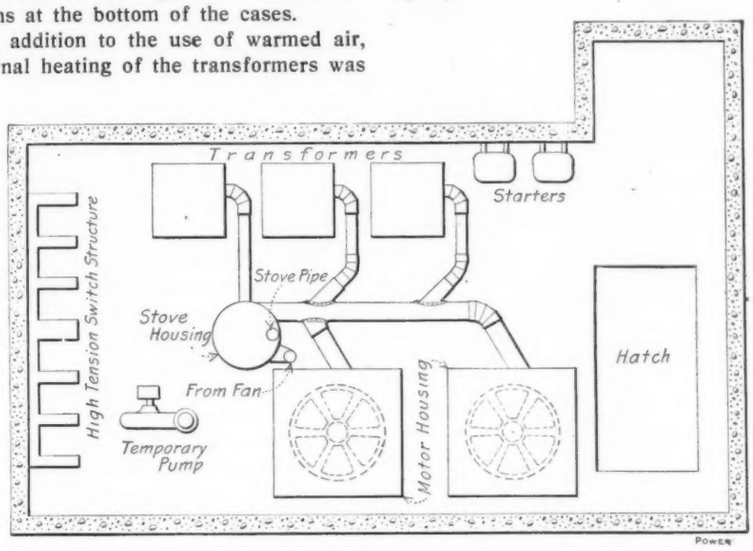


FIG. 3. PLAN VIEW OF DRYING ARRANGEMENTS

Owing to the large magnetizing current necessary for the motors and the limited capacity of the 400-volt line, it was decided to dry the motors by external heat only. These being of low voltage, it was not necessary to take such precautions as in the case of the transformers. To increase the rate of drying, a heater consisting of a few coils of No. 12 iron wire was placed in each motor housing.

A reel of cambric-insulated cable was in the sump at the time of the flood. This was unreeled and hung up in a warm room for two weeks, after which it was taken down and about 10 feet cut from each end and discarded; there being a possibility of the water having worked into the ends. The remainder was immersed in water, with the ends out, for 12 hours and then subjected to a pressure test of 22,000 volts for five minutes, which it withstood without any sign of breaking down.

Curiously enough, several barrels of transformer oil were unaffected by the flood. These barrels had been opened and, consequently, the seals were broken, although the plugs were screwed in tightly.

Readings were taken from time to time with a portable testing set to determine the insulation resistance of the transformers and motors. These showed at the start a "leakage" of 15 volts (on a 110-volt circuit) between the primary and secondary windings of the transformers or between either winding and the ground, and 90 volts between the motor windings and the ground. After drying out for five weeks the transformer readings were brought up to 1700 megohms and the motors to a point almost as good.

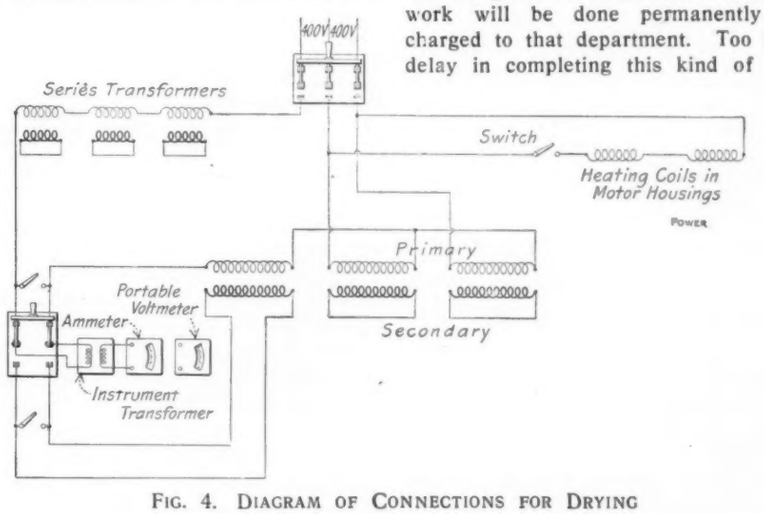


FIG. 4. DIAGRAM OF CONNECTIONS FOR DRYING

It was then deemed safe to subject the transformers and wiring to a high-tension test. This was accordingly done by means of a portable testing transformer. With the exception of three of the small series transformers, which had to be retaped, the high-tension side of the equipment withstood 22,000 volts for five minutes and was pronounced safe for continuous operation at 11,000 volts.

The system has been in operation ever since without giving the slightest trouble.

Installing Electric Motors

BY THOMAS H. WATSON

Too often a motor is situated in an almost inaccessible place, with the result that in order to give it the attention it should have the motor must be removed. In the case of exhaust fans, the motors are often placed high up in a skylight, and can be reached only by a long extension ladder, with no means of support while working on it, so that the attendant can hardly be blamed for neglecting them.

Steam traps and electric motors make poor companions, but you sometimes find them placed side by side. To place a motor beside a flue or chimney taking the hot gases from furnaces is about as bad, but it is also done. Motors driving bilge pumps that remove hot drainage when placed over the hotwell soon become grounded from the steam condensing on them and give a lot of commutator trouble.

When putting in motors temporarily some kind of follow-up method should be used to keep track of the work, and notice should be given the person in charge of the department in which the motor is placed that just so much time will be given for trial, after which the work will be done permanently and charged to that department. Too much delay in completing this kind of work leads to a lot of broken-down wiring and trouble.

Where adjustable-speed motors are installed care should be taken in placing the resistances; they sometimes run very hot and might cause a fire. The controller should be so constructed that the sliding shoes come squarely over the stationary buttons and leave no possibility for poor contact and consequent burning.

Frothing of Storage Batteries

By John S. Leese

Owners of internal-combustion engines fitted with storage-battery ignition are not always electricans, and they are frequently bothered by the frothing of the batteries while they are being recharged. When this takes place the battery loses its strength rapidly and its voltage drops, whether it is being used or not. Chronic frothing is generally due to what is known as sulphating of the plates, meaning the production of little patches of lead sulphate on the plates. The rate at which a cell can be charged and discharged without injuring it depends upon the effective areas of the plates, and upon the rate of the chemical action by which the electrical energy of the cell is set up. If, then, the area of the plates is diminished by lead sulphate and the charging current is kept the same, it cannot all be used in producing chemical action on the plates, and the excess acts upon the electrolyte, which it decomposes and causes to give off oxygen and hydrogen, which is called frothing.

The most common cause of sulphating is excessively heavy overcharging, which causes injurious chemical action to take place. The best cure is to give the sulphated battery long and frequent charges at not more than half the usual charging rate, with the addition to the electrolyte of a solution of caustic soda and water in the proportions of one part of caustic soda to five of water. The amount of this solution to be added to the electrolyte is one-fifth of a drachm to one pint of electrolyte.

Though this is undoubtedly the best way of curing sulphated battery plates, it is a long process, and operators must not be dismayed if there is but little improvement to be seen after the first one or two recharges.

Electrocuted by Current at 250 Volts

An employee at the Orr Worsted Mill, Piqua, O., was instantly killed while mopping up the floor in the vicinity of the switchboard. His right shoulder came in contact with the switch and he was unable to move.

The fireman saw that the current was going through the man's body and immediately shut off the engine, stopping the dynamo. When the current ceased, the man toppled to the floor and expired after two or three gasps.

Current at the mills is reported by the daily press to be generated at 250 volts. This is not a high enough voltage to ordinarily cause death, but the man had hold of a wire-wrapped hose, forming a good ground, and more than likely he was badly burned by the current.

LETTERS

Kerosene for Lubricating Brushes

I have read occasionally in the columns of Power suggestions for using kerosene as a lubricant on the commutator and brushes of dynamos, so I tried it. The brushes were soaked in the kerosene and, after wiping them well with a piece of cloth, were allowed to be dry. This gave a good deal of success on a 110-volt dynamo carrying about 65 amperes. The same treatment was accorded to the brushes of another dynamo working at 450 volts and carrying about 400 amperes, but in this case it was not a success. On the contrary, the brushes became more noisy and sparking seemed to increase. An effort was made to find out if there was any other cause of the poor working, but none could be found. To make it surer, the brushes soaked in the kerosene and dried were replaced by other carbon brushes, which had not been soaked in the oil; after a while the sparking ceased and the chattering decreased. Both of the dynamos were in the same room, both were compound-wound machines and both had carbon brushes. The smaller dynamo is used for lighting service and the larger one supplies current to motors driving machinery.

Will someone tell me the cause of this apparently paradoxical experience? Is it probable that the higher voltage and greater current-carrying capacity can have anything to do with the difference in the results, and if so, where is the limit?

Manilal K. Desai.

Ahmedabad, India.

Grooving Commutator Mica

We have in our station a 110-horsepower direct-current motor, belt-connected to a 150-kilowatt alternating-current generator. This motor used to give a great deal of trouble by sparking at the brushes, pitting the commutator and burning it black in spots, about the width of five or six bars, leaving about that many clear in between the black places all around the commutator.

Not long ago I saw an article in Power which described the experience of another reader with the same trouble, which he had cured by cutting the mica down between the bars, so I tried it. I made two hooks about like ordinary packing hooks but bent over a little more at the point so as to make them dig into the mica, and tempered them very hard. Then with the aid of a straightedge laid alongside the mica to guide the hook I found it a very easy job to cut the mica down about 1/64 inch.

This motor is started up at 7 a.m. and runs until 11 p.m., and formerly I often had to stop it two or three times during this run to sandpaper the commutator, but since cutting down the mica I have had no trouble at all and the commutator has taken on a nice chocolate color.

I think this practice is all right for high-speed machines, but would not recommend it for low-speed machines, because the carbon dust and other foreign matter that might find lodgment in the grooves would not be so readily thrown out by centrifugal force at low speeds but might stay there and cause a partial short-circuit between the commutator bars.

J. R. Wampler.

Staunton, Va.

Commutator Lubrication

The practice of using vaseline or commutator compounds to lubricate commutators, collector rings and brushes, so generally followed, has given me more trouble than the use of a clean cotton cloth saturated in gasolene; holding the cloth lightly against the collector rings or commutator does the work. I do not use grease of any kind. The brushes are all of carbon on the machines that I am running, which are alternators of the revolving-field type, running at 3600 revolutions per minute, and the usual direct-current exciter.

Charles Malone.

Harlowton, Mont.

Remedies for the Loosening of Squirrel Cage Bars

It is sometimes found difficult to make the bars of a squirrel-cage rotor stay fast, if they are soldered to the end rings. The solder between the bars and the short-circuiting rings either breaks loose, due to mechanical stresses, or is melted out by heavy secondary currents at starting, allowing the bars to fly out under the influence of centrifugal force. The breakage of solder may be prevented by bolting the bars to the end rings and thereby relieving the solder of the mechanical duty.

If the starting conditions are unusually severe, trouble from melting solder might be prevented by replacing the end rings with heavier ones; another and easier expedient is to wind over the ends of the bars heavy bands of No. 16 or 18 brass wire, according to the clearance, one band at each end of the rotor, without any insulation under it, and solder these bands to the ends of the rotor bars. If neither of these remedies is effective, there is nothing left but to relieve the heavy starting conditions.

C. J. Fuetterer.

Thomas, W. Va.

Gas Power Department

Everything worth while in the gas engine and producer industry will be treated here in a way that can be of use to practical men

A Suggested Solution of the Gas Turbine Problem

By Benjamin H. Blaisdell

The chief and seemingly insurmountable obstacle that has prevented the realization of success in the development of a practical gas turbine is the inability of constructive materials to withstand the high temperature of the working fluid without some means of cooling either the products of combustion before they enter the turbine or the parts of the turbine subject to contact with the hot gases, at the same time not materially reducing the heat available for conversion into useful work. The blades or vanes of a turbine become so intermingled, as it were, with the working fluid that water jacketing, to be effective in preventing their destruction, must absorb the greater part of the heat of combustion, the turbine thereby becoming a more efficient hot-water or steam-making machine than a machine for developing mechanical power.

If, on the other hand, steam or water or both simultaneously be injected into the hot gases before admission to the turbine, the ratio of water vapor to gas would necessarily be so great, in order to reduce the temperature to a safe degree, as to practically change the working fluid to the characteristics of superheated steam with its comparatively low heat range and high latent heat losses in the exhaust aggravated by the impracticability of condensing.

Divergent nozzles have been employed in which the gases are expanded to the pressure of the atmosphere before impinging on the buckets; still the temperature of the issuing gas is too great for the turbine parts to withstand it.

The restriction, thus far, of the gas turbine to the constant-pressure cycle is another hindrance to high thermal efficiency for the same reasons that this cycle in reciprocating gas-engine practice has been proved to be of less practical efficiency than the constant-volume cycle employed in all modern gas engines. In the constant-pressure cycle not only much higher initial compression is required to obtain a given mean effective pressure but the operation of compressing the gases must be accomplished in a separate cylinder or machine from that in which the power is generated, requiring the handling of gaseous fluids in comparatively large compressors. This cannot be done efficiently by present methods on account of the loss to the jacket water of much of the heat generated by the compression of the gases and loss by radiation before the heat can be converted into power.

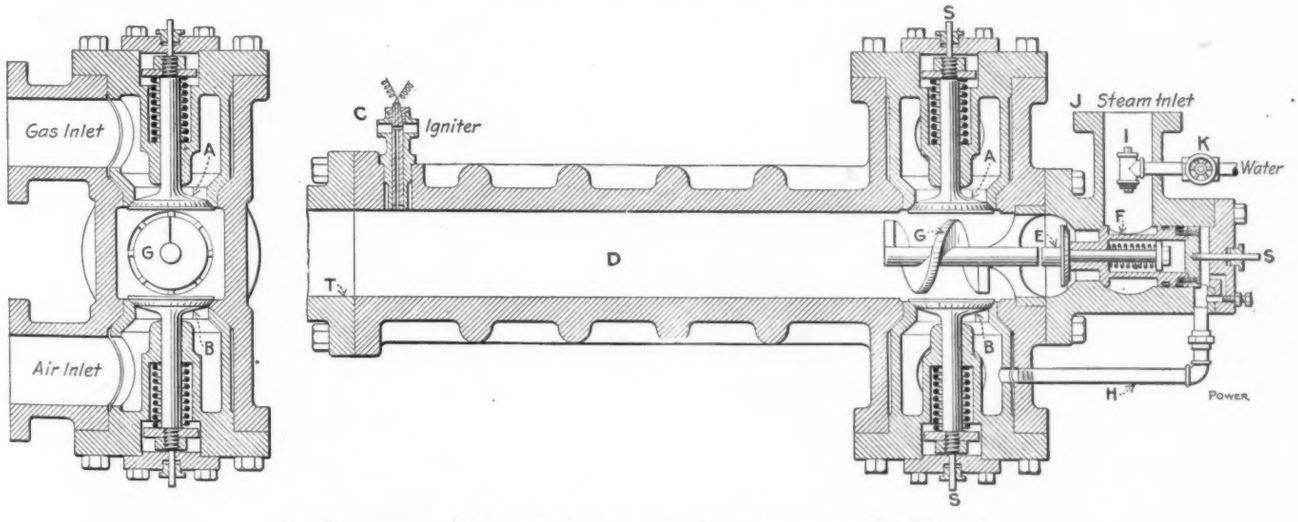
In the modern gas engine this heat loss is considerably diminished, since the events of compression and expansion occur in the same cylinder, giving little time for the heat of compression to become dissipated before it can be utilized during the working stroke.

It is the purpose of this article to describe a method or cycle which would seem to overcome the difficulties mentioned and make the gas turbine a practical success not only from a mechanical standpoint but in its efficiency of heat conversion.

The heat-absorbing or conducting power of materials is not instantaneous; they will not assume the temperature corresponding to a surrounding medium without an appreciable lapse of time.

If a piece of iron were held momentarily in a flame of, say, 3000 degrees and then plunged into a medium of a much lower temperature, say, steam at 300 or 400 degrees, this performance could be repeated indefinitely without a destructive rise in the temperature of the iron. It is just this principle which it is proposed to apply in the operation of a turbine to prevent the hot gases from raising the temperature of the interior parts above the safe limit.

The products of combustion will enter and pass through the turbine during the period of one explosion and expansion unmixed with any cooling fluid to lower the thermal efficiency, then will follow a period of steam admission at high pressure and in a moist state for the purpose primarily of lowering the slight heat rise due to the previous passage of the



Mr. Blaisdell's Proposed Combustion Chamber for a Gas Turbine

hot gases; incidentally it will augment by its expansion the power output of the machine. These two events will follow each other continuously and no water jacketing of the turbine will be required; in fact, the turbine may be covered with heat-insulating material, thereby eliminating entirely the present heat loss to the cooling water which is unavoidable in all gas engines.

To generate the steam for cooling it is proposed to utilize what in present practice is waste heat. If the exhaust gases be passed through a properly designed boiler, steam may be generated at a pressure of 200 pounds for passage through the combustion chamber and turbine, thereby reducing the temperature of the exhaust to 500 degrees Fahrenheit, or even less, provided the water vapor in the exhaust is maintained in a superheated state to prevent the formation of sulphuric acid which might endanger the metal of the boiler.

Furthermore, the heat of compression of the gases can be utilized by passing the boiler-feed water through the compressor jackets and coolers. The combustion chamber may or may not be water jacketed. For the best thermal efficiency it would be better to insulate the combustion chamber and depend on the steam for preventing a serious temperature rise. Any degree of cooling may be obtained by supersaturating the steam to a greater or less degree.

It is proposed to employ a combustion chamber which completely fills with the combustible mixture before ignition, and when combustion occurs it is so rapid as to be practically an explosion, the maximum pressure reached depending on the degree of compression and the heat value of the mixture used.

The combustion chamber which I propose to use will be elongated in shape and will be directly connected to the inlet of the turbine. The igniter will be located at the extreme end nearest the turbine and the admission valves for air, fuel and scavenging steam will be arranged in a chest at the other end of the chamber. All the valves will open and close automatically at the proper instants in the cycle by the fluid pressures employed and not by valve-gear mechanism. By this automatic operation the minimum pressure at the turbine inlet or nozzle will be practically that of the fuel and air supply, and the ignition cannot take place until the chamber is completely full of combustible mixture. The admission valves close at the instant of explosion, and steam is admitted alone during the period succeeding the explosion and previous to a fresh charge of air and fuel. It would be impossible to perform this cycle with equal results by valves or ports operated or controlled by mechanism driven from the turbine shaft.

The construction and operation of the apparatus will be clearer by reference to the accompanying longitudinal and cross-sections of the combustion chamber, where *A* is the admission valve for fuel; *B*, the air inlet valve; *E*, the main valve and *F* an auxiliary valve for admitting steam to the chamber from the pipe *J* which connects with the boiler operated on waste heat. At *I* is a water-inlet spray with a valve *K* for regulating the supply to give the steam the supersaturation desired for cooling; *C* is a "continuous" igniter which may be an electric hot wire, hot tube or a bunsen flame, and *T* is the inlet nozzle of the turbine. A small pipe *H* equalizes the air pressure back of the auxiliary steam valve *F* with the supply pressure; *G* is a spiral baffle provided to give the gas and air a rotary motion in their passage through the combustion chamber and thereby insure their thorough mixture. Small stems *S* are secured to the various valves and project outward through stuffing boxes for indicating the opening and closing of the valves.

The operation is expected to be as follows: Air and fuel enter the combustion chamber through their respective valves in proportions regulated to form the most effective mixture and when the volume admitted is just sufficient to reach the igniter, the mixture is exploded and the rapid rise in pressure instantly closes the inwardly opening admission valve, making the path through the turbine the only exit for the products of combustion.

When the pressure due to the explosion is reduced by expansion to practically that of the steam supply, the main steam valve opens, allowing the steam in a supersaturated state to enter the chamber, driving out the remaining burnt gases and in its passage through the turbine, cooling it and generating power.

The auxiliary valve is so balanced that when the air pressure back of the piston is equal to the steam pressure between the two disks, it closes; in fact, the steam flow assists in its closure, thus assuring the discontinuance of steam admission before the opening of the air and fuel valves again. This completes one cycle and the performance is automatically repeated as long as the conditions necessary for its operation continue.

The proper proportioning of the mixture may be obtained by carrying the air pressure a certain degree higher than that of the fuel gas. This excess pressure of the air would further be beneficial to the operations in the combustion chamber by causing the air valve to open a trifle earlier and close a trifle later than the gas valve, thereby admitting a small quantity of air to act as a buffer between the combustible charge and the scavenging steam.

In order to use liquid fuel a vaporizer or carbureter would be connected between the air-supply main and the air-inlet valve, dispensing with the independent fuel valve shown in the drawing.

The advantages contributing to high efficiency which a turbine operated as just described would have over that of the present type of reciprocating gas engine are manifold.

The air and gas being compressed prior to admission to the combustion chamber, their temperature can be reduced by means of water jackets, making the fluid denser for a given volume and pressure, thereby yielding a higher maximum pressure of explosion with a given maximum temperature.

The cooler mixture would make it possible to use a higher compression and a greater percentage of hydrogen in the gas without the danger of preignition.

The heat absorbed by the water in compressor jackets can be utilized by feeding this water to the waste-heat boiler.

There being no rubbing surfaces in contact with hot gases, the interior parts of the turbine can attain safely a higher temperature.

The maximum pressure of combustion can be obtained without the necessity of water jacketing, resulting in a less rapid fall during expansion and consequently a higher mean effective pressure.

The scavenging steam will recover much of the heat delivered by the hot gases to the turbine walls and convert it into power in the later turbine stages.

The working fluid flowing always in the one direction through a smooth combustion chamber insures a complete scavenging of the chamber prior to admitting a fresh charge of air and fuel.

Expansion can be carried to atmospheric pressure, which is not practical in a reciprocating engine.

A large part of the heat now wasted in the exhaust gases and cylinder jackets is utilized in the machine itself.

The slightly erosive action of the supersaturated steam will prevent the nozzles and buckets from becoming coated with carbon or scale.

With all these advantageous conditions favoring the turbine unit we might expect the heat balance as compared with that of the gas engine to be about as stated in the following table:

	Gas Turbine	Gas Engine
Percentage of the total heat of the fuel gas lost in the exhaust gases	17	35
Percentage lost in radiation and cooling water	8	25
Percentage lost internally, including mechanical friction and inefficiency of compression, transmission, combustion and expulsion of the gases	25	15
Total losses	50	75
Percentage converted into brake-power	50	25

Readers with Something to Say

Practical information from the man on the job. A letter good enough to print here will be paid for. Ideas, not mere words, wanted

Recollections of a Boiler Inspector

It appears remarkable in this day of high-class engineering papers, by means of which even the most isolated man has access to a higher knowledge of engineering, that one finds so many ignorant engineers. Following are related a few instances that have come under my observation, which illustrate the criminal carelessness and ignorance of men having charge of steam boilers:

Within the past 18 months I have found five boilers that were offered for insurance, which had stop valves between the boiler and the safety valves, and in only one of these cases did anyone appear to realize that anything was wrong. The engineer said that he knew the stop valve should not be in such a place, but had not had the time to make the change. In each case, when the seriousness of the conditions were pointed out, a change was made without delay. In one instance the manager, when advised of the dangerous arrangement, gave orders for an immediate shutdown and the boilers were not operated again until the valve had been removed.

In another plant I found a 60-inch by 14-foot horizontal return-tubular boiler being operated without a safety valve at all. This boiler was located just out of the city limits, and the negro fireman lived with his family in the factory yard, not more than 50 yards from the boiler house. He would go out about 4 o'clock a.m. and fire up, and leave the boiler until after breakfast, but he said that he never found more than 110 pounds of steam showing in the gage when he returned.

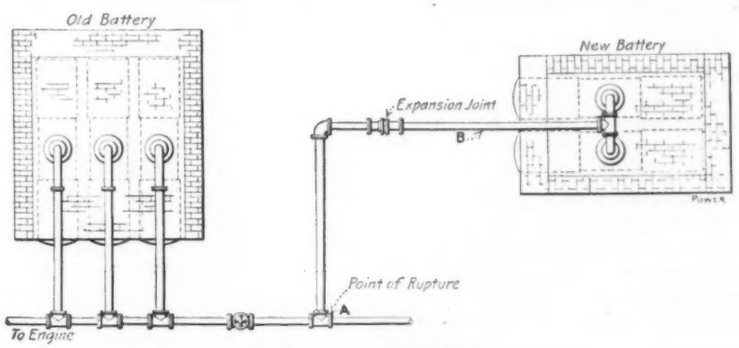
A new 54-inch return-tubular boiler had been set up. In shipping the boiler from the shop the steam gage was lost and the negro fireman went over to a neighboring gin house, found an old steam gage and proceeded to put it on his boiler and then fired up. The boiler was equipped with a ball and lever safety valve. When the steam gage showed 60 pounds the safety valve blew. This did not suit the fireman, as he claimed it was necessary to have a steam pressure of 80 pounds to operate the engine. He, therefore, removed the lever and taking it to a forge had 12 inches additional length welded to it and then moved the weight out to the end of the lever. He fired up and when the needle on the steam gage registered 60 pounds the safety valve blew again. He then made another visit to the gin plant, got the weight belonging to the old boiler-safety valve and hung it on the lever in addition to the weight already in place. The next thing that happened was that the boiler exploded and came to rest about 900 feet from its starting point in the middle of a public road. The fireman, his helper and one other employee were instantly killed. The old steam gage was found and I managed to get hold of it, and upon testing it I found that the needle would go to the 60-pound mark and hang there, although I ran the pressure up to the limit of 300 pounds on my gage. This old gage had become rusted and foul and was absolutely worthless.

W. E. NORSTER.

Shreveport, La.

Improper Piping

The accompanying sketch shows how an accident happened when cutting in a new battery of boilers. Three boilers were in service, and as the requirements of the plant demanded additional steam two new boilers were installed in an adjoining shed as there was no room in the old plant. As the steam line was of considerable length an expansion joint was inserted. It was so arranged as to move parallel with the steam main, causing a strain at the joint *A*. As a consequence, when the two boilers were cut in on the old steam line the connection *A* was broken loose by the working of the pipe *B*, notwithstanding the expansion joint.



A CASE OF POOR PIPING

EDWARD T. BINNS.

Philadelphia, Penn.

Crude Repair Job

One of the most unworkmanlike repairs on a steam engine I have ever witnessed was perpetrated in one of the largest power houses on the Pacific coast a few months ago.

I was an operating engineer in the plant, and the so called repairs were made under the direction of the operating chief engineer, who had never had a day's practical training in his life.

In one portion of the power house were two horizontal, tandem-compound, surface-condensing engines, each having cylinders of 18 and 26x16 inches. Steam was used at a pressure of 180 pounds per square inch. Each engine was direct connected to two generators, each of 200 amperes capacity, the generators being on either side of the single crank.

It is usual to run these engines 24 hours per day and seven days a week except when the main plant is in operation, which supplies power to the machines in the manufacturing plant, therefore, one engine is always running. They are changed over on alternate days. Immediately the main plant is closed down, during meal hours or at night, both engines are required to be in operation to carry the lighting load.

Closing down the main plant at noon one day, the second engine had just taken the load, when suddenly a severe knocking occurred in one of the engines at every revolution. An examination showed that some small pieces from a broken piston ring had come between the high-

pressure piston and the cylinder cover. As the clearance was small the cylinder head was cracked and the piston rod was very much bent.

On closer inspection it was found that the engine must have been out of line, as a shoulder had worn on the bottom half of the outer end of the high-pressure cylinder, and a similar shoulder on the top half at the inner end; the neck bush was also badly worn. The same state of affairs was found in the low-pressure cylinder, except that the shoulders were worn on opposite sides halfway around at the outer and inner ends. These shoulders were from $\frac{3}{32}$ to $\frac{1}{8}$ inch high at the worst places, and tapered to nothing in the cylinder walls.

After an inspection of the cylinders the operating chief gave instructions to get pneumatic hand tools, and affixing rough 8-inch diameter emery wheels on their shafts, he proceeded to grind off the shoulders on the cylinder walls, and afterward smoothed them down with No. 0 emery cloth. It was suggested that he put a line through the cylinders to find out how much they were out of line, and then rebore them true. He stated that the engine had been lined up a year before, and that he would prevent further wear on the shoulders by increasing the width of the piston rings.

Accordingly, the recesses for the rings were turned out, and increased $\frac{1}{8}$ inch in width in each case, new rings were fitted to the pistons, new rod and neck bushes were put in, and the engine again assembled without lining up the cylinders.

When started up the engine pounded badly. Numerous alterations of the valve were made, but without silencing the noise, and recently I heard that the engine is permanently out of commission and that the operating chief has been superseded.

Seattle, Wash. JOHN CREEN.

Repairing a Steel Stack

Some seven or eight years ago a 46-foot steel extension, 40 inches in diameter, was added to a square-brick chimney which was 72 inches high and 4 feet square, internal dimensions. The stack was unlined and was made with lapped joints, as shown in the illustration. Each full section was made of two pieces of rolled sheet steel varying in thickness from about $\frac{3}{8}$ inch at the bottom to $\frac{5}{16}$ inch at the top.

Some time during the Sunday shutdown the section *A* dropped down through the steel extension, through the brick stack and landed at the base of the chimney, without doing further damage. The sheet was removed through a large door in the base of the chimney. The local boilermaker who had made the stack advised taking the stack down and putting up a new one, and a local contractor offered to build a staging about the stack and to replace the sheet for about $300.

Finally, a former sailor volunteered to carry out the work as follows: A ½-inch single-snatch block was put around the three guy wires and the rope and tackle were pulled up, on the guy wires to the top, as shown. The more times the rope was wrapped around the guy wires the easier it traveled.

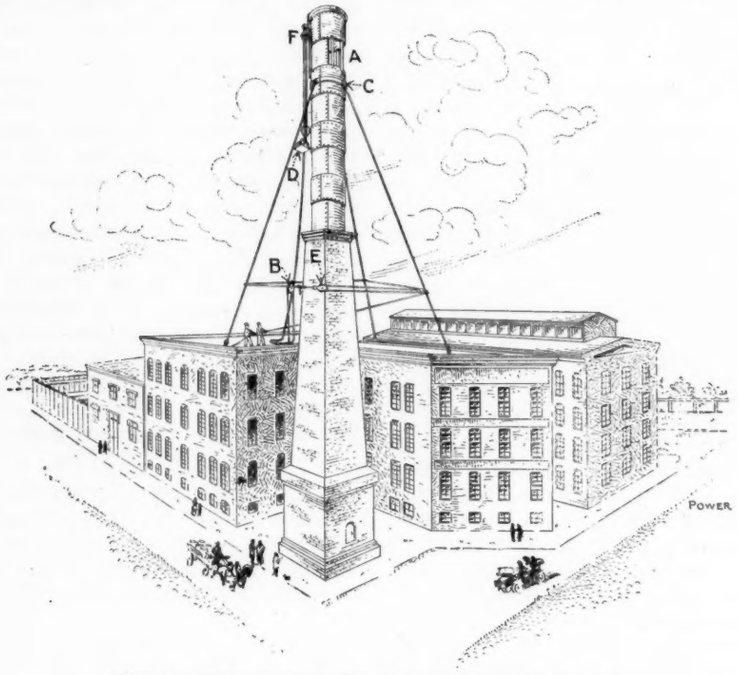
Before the single-snatch block and rope were up out of easy reach, a ½-inch double-snatch block was hooked over one section, as shown at *B*, and traveled up the guy wires with the single-snatch block ropes *E*. When the ropes and tackle had reached the point *C* the tag end of the single-snatch block was secured. The chair *D* was then hooked to the lower end of the snatch block *B*. The sailor then took a pole, hooked the snatch block *F* into the ring and was hoisted in the chair to the point *C*, when he hooked the reach pole over the top of the stack. He was then lowered and again mounting a chair, similar to the first, attached to the lower end of the snatch block *F*, he hoisted himself to a hight where an examination could be made of the sections adjacent to the section which had dropped out. Suitable measurements were taken and then three ½x1-inch bars of iron, 5 feet long, were bolted on the section which had dropped out. This was rolled and straightened out and bolted in place, leaving the bars in for extra strength. It was found that the stack was hardly sprung and there was no difficulty in replacing the original section. The accident was caused by an imperfect heading of the rivets.

While the tackle was in place the stack was given one coat of asphaltum paint. A small snatch block was left at the top of the stack through which a No. 10 weatherproof-covered wire was passed and attached to the roof below. With the small snatch block a reach pole carrying a ½-inch double-snatch block might at any time be raised and hooked over the top of the stack.

A small snatch block should always be left at the top of steel stacks or a ladder attached to the stack for convenience in painting or repairing. A steel stack should be repainted about every two years.

ROBERT E. NEWCOMB.

Holyoke, Mass.



METHOD EMPLOYED IN REPAIRING DAMAGED STACK

Loose Crank Pin

Some time ago I had a loose crank pin, the actions of which I have not yet been able to understand. The pin was on a 16x36-inch engine, running over at 100 revolutions per minute and was oiled by means of a centrifugal oiler. This pin apparently turned in the disk and screwed the centrifugal oiler out of the pin. I screwed the centrifugal oiler back into the pin and set the check nut up tight, but in a few minutes it came out again. This was repeated two or three times and I was forced to put in a new pin, forcing it in with a screw press after boring out the hole in the disk. I fail to understand why the oiler was screwed out of the pin. If the crank pin turned in the disk it must have been stationary in the crank-pin brasses and this would have caused the pin to rotate in the disk toward the left or in a direction to unscrew a right-hand thread. The centrifugal oiler may be likened to another

crank on the other end of the pin and if the pin unscrewed from the disk it should screw onto the oiler at the other end of the pin. This oiler was of the common size with a ½-inch right-hand pipe thread.

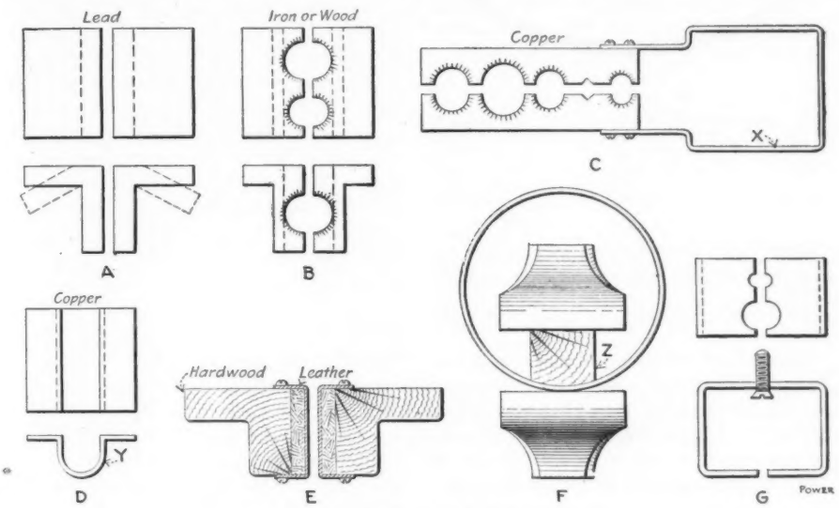
Who has an explanation?

L. A. FITTS.

Fitchburg, Mass.

Vise Clamps

In the accompanying illustrations are shown several designs of vise clamps. At *A* is shown a lead clamp made out of a piece of sheet lead folded to the desired thickness, and bent, as shown by the dotted lines, to conform to the shape of the vise jaws.



TYPES OF VISE CLAMPS

An iron clamp, shown at *B*, is used to hold iron pipes or bars in a bench vise. The designs of both side and top clamp are illustrated.

Wooden clamps of this description are valuable for holding a light tube. If the tube is lacquered it should be protected by wrapping a piece of cloth or felt around it before inserting it in the wood clamp.

What might be called a spring clamp is shown at *C*, and is generally used for holding screws, tubes, etc.; the clamp should be made of copper so that the threads of the bolts, etc., will not be damaged. A spring *X* is attached to the clamp, but a bench vise is necessary to obtain a suitable holding grip.

The clamp shown at *D* is generally made of copper, the vise jaws gripping it at *Y*.

A pair of hardwood clamps are shown at *E*, the face of each of which has a piece of felt attached, over which a covering of soft leather is fastened by means of screws or nails to the wood. They are used in holding delicate metal work requiring only moderate pressure.

A method of holding a piston ring in a vise is shown at *F*. The wooden block *Z* is used with the ring in the vise jaws.

A method of holding short screws or screw-head bolts without injury is shown at *G*.

JAMES E. NOBLE.

Toronto, Can.

Engine Runs with Steam Valves Closed

I have a 16x32-inch Atlas Corliss engine which will run at slow speed when both admission valves are closed. I have just had the valve seats rebored, new valves fitted and set with the aid of an indicator, and it runs faster now with the admission valves closed than it did before reboring.

I know of a 20x36-inch engine of the same type and make that does the same trick, although it has only been run six months. Perhaps the valves are not set right or the engine is defective. No matter how we set the admission valves the engine acts the same.

I would like to hear from other engineers having charge of this make of engine.

O. LANTZ.

Archbold, O.

Crosshead Pins

I have long contended that the crosshead pin of a steam engine should be made larger than most engine builders make them.

The friction might be a little greater, but wear which creates a knock and necessitates frequent adjusting would not be so great if there were more surface over which to distribute the wear.

The crosshead pin would not be too large if made the same size as the crank pin, although I have seen crank pins on side-crank engines that were smaller than the crosshead pin should have been.

LLOYD V. BEETS.

Nashville, Tenn.

Graduating a Safety Valve Lever

My experience has been that but very few lever-safety valves are correctly graduated.

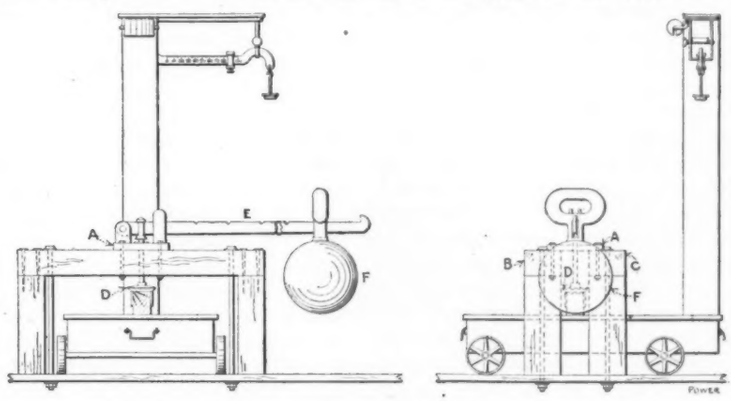
The illustration shows a 3-inch safety valve on a framework and a platform scale. First, I remove the cover from the valve body and bolt the cover *A* to the timbers *B* and *C*. The hight of the valve *D* is so adjusted that the lever *E* will be in a perfectly horizontal position. Then the weight *F* is placed at the 100 notch. The diameter of the valve being 3 inches, its area would be

$3 \times 3 \times 0.7854 = 7.0686$ *square inches.*

The total steam pressure necessary to raise the valve when the weight *F* is at the 100-pound notch is

$7.0686 \times 100 = 706.86 = 707$ *pounds.*

If the scale balances when set at 707 pounds, the lever has been graduated correctly. If it does not balance at this point, shift the weight slightly to the right or left until the scale does balance. Mark this point on the lever and it will be the correct 100-pound notch.



SAFETY VALVE ON FRAME OVER PLATFORM SCALE

In the same manner test the graduations of the other points on the lever with the scale, being careful to mark each point correctly. Of course, it is necessary that the scale be in good condition and properly balanced.

Salem, Va. E. L. MORRIS.

Questions Before the House

Comment, criticism, suggestions and debate upon various articles, letters and editorials which have appeared in previous issues

Available Heat to Steam Boilers

I have read with interest the editorial in the July 18 issue upon Mr. Morley's treatment of the available heat to steam boilers. Mr. Morley shows clearly that some of the heat is absorbed by the moisture in the coal and is lost, this moisture passing into the flue as highly superheated steam. He suggests that the heat necessary to evaporate and superheat this moisture should be deducted from the heat value of the dry coal, to obtain the heat available to evaporate the water. While his observations as to moisture in coal are true, yet there appear to be a number of objections to his proposed change in the method of computing the efficiency of the boiler.

Moisture in coal, like ash, is an inert, noncombustible substance. Both the ash and the moisture absorb heat and carry it, the one into the ashpit, the other into the stack. If the heat lost by the moisture is to be deducted from the available heat, then, to be consistent, the heat lost by the discharge of hot ashes from the furnace should likewise be deducted.

The difference indicated between the two methods of computing boiler efficiency is small. While it is true that the moisture content of coal may vary considerably from time to time, especially with coal stored in the open, the difference in efficiency that may be produced by other conditions, such as altering the proportion of air to fuel, may greatly exceed that due to moisture.

In a series of six tests made by students under the direction of the writer during last spring, and extending over a period of more than two months, the moisture in the coal was found to vary between 8.4 and 12 per cent. The same kind of coal was used, and it was stored under shelter. This does not indicate that a very large variation in moisture is to be anticipated when using coal from the same source. Furthermore, a complete report of a boiler trial includes a proximate analysis of the coal, so that in comparing boiler performances, any marked change in the composition of the fuel may be noted.

A certain amount of heat is inevitably lost through the discharge of hot gases to the stack. Even if these gases were cooled to the temperature of the steam before leaving the heating surfaces, their temperature would still be far above that of the air in the fire room. It is not the fault of the boiler that this heat is thus unavoidably lost, yet no one would think of estimating its amount and deducting it from the heat in the fuel to get the heat available for evaporating water.

A. SCOTT.

Columbia, Mo.

Turbine Accident at Riverton

I was much interested in the account of the turbine wreck at the power plant of the Illinois Traction Company, at Riverton, which was published in the August 8 issue.

While the cause of the accident probably never will be known, a few conjectures on the probable cause may not be amiss, as that is as near as it is possible to get to the facts. According to the report, the machine was supposed to be running at about half speed, and, if such was the case, the wreck was evidently due to other causes than overspeed, yet it may have been a case of overspeed after all. If the repairs that were made consisted of only an overhauling of the bearings, governor, and an internal inspection of the steam end, no change could possibly have been made that would have thrown the machine enough out of balance to wreck it. It is a fact that if one of this type of machine is shut down and allowed to cool off until quite cold and then started up in a hurry, it will vibrate very badly. I have had the pleasure of overhauling a few of these machines, and have had this same vibration occur. I always found it due to the fact that too much load is thrown on the machine at once, and that one side of the machine heated up faster than the other, and put an extra side strain on the shaft. The only thing I ever did in such a case was to throw the load off and run the machine empty for an hour or so, letting it come to a uniform temperature and then applied the load gradually and the vibration always disappeared.

In the description of the Riverton wreck it was stated that the governor was overhauled. Now, it is well known among turbine operators that an adjustment of about ¼ inch on the stem of the pilot valve of the hydraulic governor may cause one or more valves to stay open, and that the machine cannot run fast enough to cause the governor to close them. If one were watching the valves to see them close and saw two or three still open, he would naturally think that the turbine was not up to full speed. I worked about turbines for two years before I got on to this pilot-valve stem stunt. Only lately I had a turbine trip the throttle twice by the emergency device. The first time I thought it was the fault of the hook that holds the weight up, as it had been giving some trouble, but when I tried to hook it up again I found that the emergency ring was too far out, so I knew it was caused by overspeed. I opened the throttle again, and before I discovered which valve was sticking, the emergency mechanism went out again. Now, there was no indication of the machine running over normal speed, which was 750 revolutions, but it had crept up to 850, or enough to trip the emergency valve. This overspeeding was caused by only one valve out of 16 remaining open. It is very important to have the emergency stop always in working order, for if it came to a "showdown" with the old type of throttle, one could not close it by hand soon enough to save the machine if some of the valves hung up. It is absolutely imperative that each time the machine is shut down, it be done by tripping the throttle. The safety device should be tested very frequently, and always after working on the carbon packing, as on some machines it is necessary to disarrange part of the safety mechanism when working on the packing.

This testing is done by having someone hold a tachometer on the shaft while some one else holds a valve or two open and another stands ready to trip the throttle if necessary. One may readily see from the foregoing that it is possible for a machine to attain a dangerous speed in a short time while everything is apparently all right.

I hardly think the burst wheel was damaged by tools being left in the machine, for they could hardly get in as far as the hub of the wheel or diaphragm; besides, the rotating of the machine would tend to throw the tools out to the bucket edge of the machine, and the noise set up under these conditions would plainly indicate that something was wrong inside. On the other hand,

if something were caught up next to the hub, the vibration necessarily set up would call attention to it. The broken condition of the hub of the diaphragm does not necessarily denote that anything was caught there, for it is self-evident that even after the wheel had broken, due to the terrific speed at which it was running, if such was the case, the diaphragm would tend to hold the wheel in the casing until such time as the hole in the diaphragm became large enough to permit the pieces to fly out. Whether the step-bearing oil pressure failed would, I think, have been no cause for such a wreck, for the symptoms of a step-bearing oil-pressure failure are not to be mistaken. When the pressure fails the oil remaining around the step and guide bearing takes on various colors and temperatures, mostly black, and smoke rolls out of the vent pipes, creating a very noticeable odor, the machine vibrates badly, and in the course of a half minute or so one may hear the shroudings begin to rub. Generally, if one gets the throttle closed at this time the material damage is usually confined to the step and guide bearings, the damage to the step being due to natural (or unnatural) causes, and the guide bearings being damaged owing to the fact that all the cuttings from the step have to pass through the guide bearing if there is any oil circulating at all.

Of course, there will always remain the theory that this wreck was caused by a flaw in the wheel. But it does not seem possible that the wheel would run for a number of years and then give way under half speed. I am inclined to believe the cause of the disaster was overspeed.

K. C. Jones.

Reno, Nev.

An Engineer's Experience

I went through an experience in the winter of 1907-8 that I do not care to repeat. The plant in which I was employed contained about 1500 horsepower of water-tube boilers. As business was poor and orders were few and far between, it was decided to shut down for two weeks during January. In order to save expense the superintendent ordered all of the fires drawn and the boilers and pumps drained. The pipes were drained as far as possible, but there were some pipes that could not be drained. I protested that fire should be kept under one boiler at least. The temperature went down to 10 degrees below zero, and stayed there a week.

One afternoon I was called out at four o'clock with the temperature at zero and no light to work by but torches, and told to secure some men and get the plant ready to run the next day. "Hunkies" were the only laborers I could get to carry water to fill the boiler which, owing to the bad weather, took nearly all night. I had asked permission to bring a locomotive crane, which was fired up, around to the boiler room and run a temporary line from it to a small pump and use that to fill the boiler, but the superintendent had told me to "fill her by hand." We filled it by 2 a.m., and, after knocking some of the ice off the manhole, we got the plates in. With the water level just up to the manholes there was just 2 inches of water showing in the glass. Firing up under the circumstances seemed dangerous, and I hated to try it, but I was told that if I did not want to do it there were a dozen engineers in town who would. So I started the fire with 2 inches of water and everything frozen solid. We had previously built a fire under the feed pump to thaw it out, and at about 3.30 we had 40 pounds steam pressure with which we tried to start the pump to fill another boiler. The pump would not start, however, although I tried to thaw it out with steam. All the time the water level was getting lower. We had to cut in 110 feet of 12-inch header to get steam to the pump, and this condensed a great deal of steam; besides, the blowoff may have leaked some.

In a short time we heard a tube pop and then another, and by the time we had pulled what little fire there was left the rest were burned. The result was that the next week a boilermaker put in 120 new tubes and five headers which, with other repairs, cost about $1000.

I certainly was mad and discouraged that morning, and my clothes were frozen stiff. We tried the same trick with the No. 2 boiler. While the laborers were filling it I examined the pump and found one cylinder about half full of ice. Evidently the bleeder had become plugged up with scale and the water did not all drain out.

We got No. 2 full and a fire started when the tube caps began to leak. The fire was pulled in time to prevent burning this boiler. The caps on both ends of No. 3 were then taken off, cleaned and replaced. While this was being done the superintendent consented to use the locomotive crane to furnish steam to the small pump and to thaw out the feed pump. After a while we got the pump to pull water from the pond and filled No. 3 boiler. After we got this boiler on the line things began to look prosperous, and after 40 hours of steady hustling and no sleep we got the plant started with three boilers. The total cost of putting the plant in operation, including the damage to No. 1 boiler, was about $1200.

I was exonorated of all blame for the burned boiler by the owners, but that did not help matters any.

This trouble could have been avoided in three ways. Connecting the city water to the main water line would have cost about $25. The locomotive crane could have been used in the first place, or one boiler could have been kept fired up for about $35. But they would not listen to the advice of their engineer.

This is only one case of several that I could cite in which the engineer was not allowed to use his best judgment.

Mr. Beets, in the August 8 issue, asks who is to blame when plant owners are not more informed regarding dangerous conditions. He also asks why should an engineer be forced to take questionable orders from a boss, who is ignorant, etc.? He should not, but the boss says, "That boiler belongs to me, and the inspector says it is safe. If you don't want to do as I say you can quit and someone else will do it." When a man has a family to care for and no money ahead, what can he do but obey orders? The engineer can help matters by showing his employer that he is an expert in his line and not a common laborer, and by refusing to work for a firm that will not recognize him and pay him as such.

J. Case.

Hyattsville, Md.

Massachusetts License Laws and Examiners

After relieving himself of a heavy burden "touchin' on an' appertainin' to" the Massachusetts license law and examiners, J. A. Levy, in the August 1 issue, says that he knows that his sentiments regarding the Massachusetts license system are held by men all over the State.

Their name is legion who are fond of expressing those sentiments, they are practical men only in their own estimation, but when weighed in the balance they are found wanting. The term practical engineer is indefinite, its meaning depending upon the character of the person using it. While there are a few men of no education who are able to keep a plant in good repair and economical operation, there are many who seem to think that a practical engineer is one ignorant of the philosophy and mathematics of steam engineering but who has been permitted to "bluff" a job. There is also a type whose practical knowledge consists in knowing how to "queer" a plant when about to be discharged, so as to make his successor sick of his job. One of these was discovered by a man who, on beginning his duties in a hotel, found the steam-heating and hot-water systems and electric-elevator service out of commission since the "practical man" had left. When the successor closed the blowoff valve on a hot-water heater he soon had steam and hot water to give away.

The fault with the electric elevator was that the commutator was varnished and there was an open circuit in the solenoid which resisted the brake spring. Another

"practical engineer," when about to be discharged, set the steam gage ahead and the safety valve back 40 pounds, so the engine would not carry the load. Still another believed in doing his dirty work before being discharged. His pump stopped for want of cylinder oil and his finish came when he wrote a note to the superintendent saying that the man in charge did not know how to set a valve.

When such people denounce the license system, their "kick" only reacts to bring to the support of the law every man who believes in qualifying before expecting to get a license. Instead of complaining that the present scope of examinations is too severe, a more thorough examination should be asked for. Suppose an applicant goes prepared to answer 2000 questions. After he has given 100 correct answers, the examiner takes it for granted that he can answer 1900 more, and issues him a first-class license.

What can be said of a case where a man goes prepared to answer only 50 questions and is given a first-class license after correctly answering 30? It could be said that he was lucky.

To eliminate the element of luck, the examiner has no recourse; but he is better able to judge the fitness of the applicant after several examinations, which work to the advantage of the applicant in remembering what he has learned.

If examinations are to be so slack that a would-be engineer does not have to remember anything, then the State might as well require an applicant for a license to go before an examiner and give a song-and-dance.

V. J. IRONSIDE.

Boston, Mass.

J. A. Levy's letter in the August 1 issue under the above heading interested me considerably. In steam engineering today so much theory and practice go hand in hand that the engineer who fails to keep pace with the times must sooner or later be relegated to a more humble position in favor of one who has both the necessary practical experience and theoretical training.

Mr. Levy cites a number of questions which were put by the examiners to an applicant for a second-class license who failed to pass. Mr. Levy argues that the questions were unfair. Personally, I do not see anything particularly improper in the questions quoted. I do not think that the applicant was rejected solely through his failure to answer the questions. It is possible that there was something else which the applicant did not disclose for fear of incriminating himself.

Mr. Levy further states that when men who have had charge of large power plants for years candidly make the humiliating statement that under the present conditions they could not get a third-class license, there is something wrong. It is my belief that if those engineers would take down their dust-covered engineering books and magazines and get busy with them that "something wrong" would soon disappear.

There is a type of engineer who never has time, so it is claimed, to read engineering books and uptodate magazines; but he does have plenty of time to stand on street corners with his "hammer" and "knock" and disparage the successful engineer, who improves himself whenever he can.

A. LAMARINE.

New Bedford, Mass.

Power Plant Betterment

I read the criticism of Mr. Cox in the July 4 issue of Mr. Hunt's paper, printed in the issue of May 2, and agreed so emphatically with the latter that I had to send along a few of my opinions and experiences.

While believing in the isolated plant in a great majority of instances, and admiting that there are cases where the central-station proposition is the only feasible one, I believe that in many places the latter has won out simply because of the troubles pointed out by Mr. Hunt. I am connected with a concern building steam engines and visit a good many plants in the course of a year. In selling new engines, old ones are often taken in exchange, but there is no trouble in disposing of them. No matter how bad they are, and a few of them defy description—somebody always has the money to keep them running. These are probably some of the dilapidated outfits to which Mr. Hunt referred.

Mr. Cox says most engineers possess an indicator and know how to use it to set the valves. Personally, I doubt if one engineer in ten in plants of 250 horsepower or under owns an indicator. This statement is borne out by the amount of indicating business done by the various engine-building concerns at $10 and up. The men with enough ambition to pay $50 for an indicator do not linger long in the small plants under consideration.

Our experiences in indicating engines for outside parties are often laughable. The engineer will beg for a card for his own, and when he gets it he will ask if it is a good one. One engineer, who had been running a small compound engine for about a year, wanted to know whether the drain from the receiver to the trap should be open or shut. "I do not want to waste any steam," he said, and had been running with it closed.

I know of a plant in which there are four men, two day and two night, who have all risen from yard laborers inside of 18 months. One engineer I met had never seen an indicator used without a pantograph reducing motion. To see some of the above men running an economy test and making use of a CO_2 machine would prove highly instructive. Of course, we meet a few real engineers in our rounds, but they are always with the people who pay fair wages and have uptodate plants, into which the central-station solicitor knows better than to venture.

I believe Mr. Hunt's idea of periodical visits by an expert is a fine thing, for even where the engineer is a good man and recommends improvements, "familiarity breeds contempt," and the manager follows the advice of the expert when he will not listen to his own engineer.

JOHN BAILEY.

Milwaukee, Wis.

License Agitation in Rhode Island

In POWER for May 16, I was much interested in the account of the license agitation in Rhode Island. I was not at all surprised at the opposition from those who imagine that their interests are threatened by license laws, but I was surprised to learn that a Boston man who appeared before the committee said he was opposed to the bills; that he had come to the hearing to intercede for the engineers and to save the public of Rhode Island from the troubles which had been experienced in Massachusetts. He said that the object of the bill was to legislate certain men out of positions in order that others might get them.

I had been operating and in charge of two different plants long before the license law was ever thought of in Massachusetts, and there was not very much in it in the way of money in those days. Then I went down to Rhode Island thinking to better myself, but instead I found things a good deal worse. I was sent to take charge of a large plant in a woolen mill in a small town. I reported at the office, and, after they had given me a whole lot of instructions as to what they expected of me, I asked them about money matters. They told me that they paid the last engineer $9 per week; but seeing that I had come well recommended they would pay me $10. They also said that the fireman was in charge at that time, and they could not see but what he was getting along all right. I left without even looking at their plant. That is what one may expect in a no-license State.

Some of us are kicking against license laws instead of helping them along and projecting our interests as well as that of the public. How would some of these kickers against license laws like to see laborers taken out of a trench and put in charge of steam boilers as I have seen done in a no-license State? Let them think it over.

JOHN MCINIS.

Cambridge, Mass.

Trouble with Leaking Tubes

In his letter in the July 18 issue, under the above, Mr. Reimers states that three months after the installation of a new boiler with but two sheets, double-riveted lap seams, and with two through stays below the tubes, the tubes began to leak.

I have had charge of similarly constructed boilers and never found that the stays were the cause of the tubes leaking. In one boiler that leaked the feed water entered at the front end with no internal pipe. The tubes had to be attended to every cleaning day. After an internal pipe was installed there was no more trouble from leaking tubes. In the case of an internal-furnace type of return-tubular boiler with one stay on each side of the furnace and just under the tubes, the feed water entered through the blow-off. The tubes began to leak four or five months after the boiler was installed, and within the year cracks appeared at the rivet holes on the back of the furnace. I suggested to the master mechanic that the feed pipe be changed to the center of the boiler as in the locomotive type. This was done, and not having room for an internal pipe, we made a long thread on the entering pipe and put a tee on that so that the water in entering would not strike directly on the tubes but fall between the shell and the tubes. This was a great improvement, and as the boiler was badly scaled, a new set of tubes was put in; these tubes never leaked the following year, all during which the boiler was used.

I think Mr. Reimers will find that the manner of injecting the feed water has much to do with his trouble, especially if he has no feed-water heater.

WILLIAM BEATON.

Gold Roads, Ariz.

Cylinder Lubrication

Referring to the contribution of Mr. Heck, which appeared in the July 25 issue, it is unfortunate that the results obtained did not furnish reliable data which would have enabled engineers to formulate some law as to the amount of cylinder oil necessary for an engine of a given horsepower.

The proper lubrication of steam-engine cylinders is certainly a branch of engineering in which there is a woeful lack of accurate information, and I would like to suggest that if there are any engineers who are in a position to make tests similar to those carried on at Purdue University, that they endeavor to secure data on this matter of cylinder lubrication and send it to POWER. Such information would be invaluable to the engineering fraternity at large. If such tests were made over a longer period I believe more dependable results would be obtained.

I hardly believe engineers will agree with Mr. Heck's statement that "even considering these cases of variation, it is plain that the value for frictional horsepower is practically constant for all amounts of cylinder oil fed to the engine." If this statement be true, engineers had better start to reduce the amount of cylinder oil used to the "irreducible minimum."

Some time ago the United States Government conducted some tests on a type of vertical engine used for driving light dynamos on battleships, and obtained results completely at variance with those of Mr. Heck. In these tests the following results were obtained:

Kind of Lubricant	Ounces Lubricant per Hour	Pounds Steam per Kilowatt-hour
Oil	2.2	29.7
Oil	1.0	30.9
Oil	0.5	33.1
.....	....	36.1
Kerosene	3.0	34.2
Gasolene	3.0	34.7
Soda solution	10.0	37.2

The kerosene, gasolene and soda solution were used to wash out and thoroughly remove any oil that might adhere to the cylinder wall. These tests prove conclusively that there is a reduction of over 20 per cent. in efficiency between running the cylinders thoroughly lubricated and entirely without any lubrication.

One thing which may vitiate the value of Mr. Heck's tests is that besides the possibility of the brake load varying, the friction losses of the shaft bearings, eccentric, crosshead guides, crank pins, etc., probably were not constant throughout the test. The considerable effect that these quantities might have on the results is indicated by the tests carried on some years ago by Professor Thurston. He analyzed the results of a series of tests and segregated the various friction losses in an engine. He found that in the case of a straight-line 6- by 12-inch engine with a balanced valve, 32.9 per cent. of the entire friction loss was due to the piston and rods, the remainder being consumed in the main bearings, crank pins, valve gears, etc. In the case of a straight-line 6- by 12-inch engine with unbalanced valve, 25 per cent. of the entire friction load was due to the piston and rod, the remainder as previously noted. With a 7- by 10-inch Lansing engine with locomotive valve gear, 21 per cent. of the entire friction loss was in the piston and rod, and the remainder as noted.

From these tests it is evident that during Mr. Heck's test the larger portion of the friction load may have varied enough to account for the rather erratic figures he obtained, especially in view of the fact that the bearings were lubricated by sight-feed oil cups rather than by flooded or force-feed bearing lubrication.

It might not be out of place to mention here that engineers are not so much concerned in the amount of oil necessary for bearing lubrication, for, fortunately, all of the oil used for bearings can be collected, filtered and used over and over again, as is now done in uptodate power plants. From the standpoint of economy with the cylinders, however, the case is different, as whatever oil is injected is lost beyond recall.

I would like to ask Mr. Heck how often the indicator diagrams were taken, and if there was any gradual increase in the indicated horsepower from the first diagram taken after a large drop of oil was fed and the last taken before the next drop reached the cylinder. This might throw some interesting light on the test and would give a better idea as to the value of a lubricant in cylinders. Theoretically it would seem that after a large slug of oil reached the cylinder the friction work would surely be decreased, but that on account of the oil being washed off by new steam and by the scraping action of piston rings, the cylinder walls would soon become "oil dry" and the friction work would again rise to a larger value. For instance, take the test of a straight-line engine turning at 272 revolutions per minute; when the oil was only fed at the rate of one drop every three minutes, the engine made 816 revolutions without receiving any lubrication.

If a continuous indicator were used or if the ordinary indicator diagrams were so timed that they could be referred to the time of the injection of each drop of oil, any variation in power noted, as pointed out above, would give a good line on the comparative value of hydrostatic lubrication, and some of the more recent types of force-feed lubricators now on the market.

In at least one type of the latter class of lubricators, provision is made for feeding oil drop by drop at any rate desired, but a further refinement is introduced whereby these large drops of oil are caught up and subdivided again and a small amount of oil fed into the cylinder for every stroke of the engine. Theoretically this latter principle seems to be the better, but here again actual practical information is lacking, and if this could be supplied it might enable us to entirely revise and greatly improve the present systems of cylinder lubrication.

GEORGE F. FENNO.

New York City.

Experimenting with CO_2

I was greatly interested in T. P. Williams' letter in the July 25 issue.

Will Mr. Williams kindly tell us what the uptake temperature is, whether the draft is regulated by the damper or by the furnace and ashpit doors, and at what place in the boiler setting does he obtain his samples? At the time 14 per cent. CO_2 was obtained, was any test made for CO?

JOSEPH ANDERSON.

Knowles, Cal.

Inquiries of General Interest

Questions are not answered unless accompanied by the name and address of the inquirer. This page is for you when stuck—use it

Heat Loss Due to Moisture in Wood

What is the difference in heat value between green and dry Oregon fir? In other words, what is the loss in heat for each percentage of moisture contained in the wood?

E. H.

When ordinary soft wood, such as fir, is green it contains anywhere from 25 to 50 per cent. of moisture. After drying in the air for about one year the moisture will be reduced to between 18 and 25 per cent.

The heat value of bone-dry fir is about 9000 B.t.u. per pound. Each percentage of moisture in the wood decreases the amount of heat available to the boiler by about 102 B.t.u. per pound.

Thus, green fir containing, say, 40 per cent. of moisture, would have an available heat value of

$$9000 - (102 \times 40) = 4920 \text{ } \textit{B.t.u. per pound}$$

Whereas, fairly well seasoned fir containing, say, only 20 per cent. of moisture, would have an available heat value of

$$9000 - (102 \times 20) = 6960 \text{ } \textit{B.t.u. per pound.}$$

Plunger Pump Discharge Regulation

When feeding a boiler with a power-driven plunger pump, how is the discharge controlled?

F. G. J.

On most, if not all, power-driven pumps there is a valve called the bypass which controls a passage between the suction and discharge chambers of the pump. With this valve wide open the water, instead of going out through the discharge pipe, goes into the suction chamber and is simply churned back and forth.

By partially closing the bypass the quantity of water going to the discharge pipe can be regulated to any degree up to the capacity of the pump.

Pump Dimensions for Given Delivery

An engine develops 600 horsepower on 25 pounds of steam per horsepower per hour; what size of ordinary duplex pump will be required to supply the water 25 per cent. in excess of what the engine uses?

H. D. P.

25 lb. per horsepower
600 h.p.
Min. in 1 hr, 60) 15,000 lb. per hour
250 lb. per min.
28 cu.in. per lb.
7,000 cu.in. per min.
1.25
2) 8,750 cu.in. per min., 25% excess
Piston sp'd, in. per min. 600) 4,375 cu.in. per min., each cylinder
7.46 sq.in. = area plunger

A 3-inch plunger has 7.07 square inches area, and will be large enough in view of the low piston speed used, 50 feet per minute.

Strength of Cone Seam

A tank is 48 inches in diameter and built of ¼-inch plate which has a tensile strength of 60,000 pounds per square inch. At the lower end of the tank is a cone which has a single-riveted lap seam. The rivet holes are 11/16 inch in diameter and the pitch is 2 inches. What is the strength of the seam?

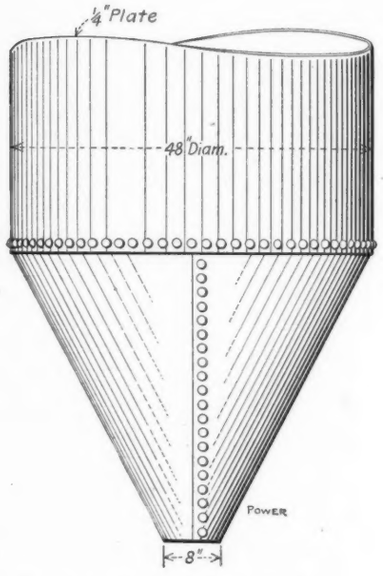
P. L.

The longitudinal joint of a cone-shaped section which is withstanding internal pressure is subjected to a varying stress. The stress at any point in such a joint will be inversely proportional to the distance of the point from the axis of the cone. It is not practicable to make a joint that would offer equivalent resistance to these varying stresses; hence, it is customary when the cone is made of one sheet to make it of the same strength as the tank itself. If several sheets are used in making up the length of the cone, the longitudinal joint of each course might be designed for strength inversely as its distance from the axis. This, however, is rarely done except in large tank work where the size makes the saving in material an object.

The strength of the seam is the strength of its weakest part, which, neglecting the crushing of the sheet in front of the rivets, must be either the tensile strength of the ligament between the rivets or the shearing strength of the rivets themselves. The strength of the portion of the plate between the rivets is

$$\frac{2 - 0.6875}{2} \times \frac{60{,}000}{4} = 9943.75 \text{ } \textit{pounds}$$

per inch of seam length.



TANK WITH CONE-SHAPED BOTTOM

Steel rivets in single shear are assumed to have a shearing strength of 42,000 pounds per square inch of cross-sectional area; the cross-sectional area of a rivet 11/16 inch in diameter is 0.37122 square inch, and the strength of the rivets is

$$\frac{0.37122 \times 42{,}000}{2} = 7795.62 \text{ } \textit{pounds}$$

per inch of seam length, which, being weaker than the ligament, is the strength of the seam.

While the seam is of equal strength throughout its length the internal pressure per square inch required to shear the rivets will vary directly as the diameter of the cone. For instance, on a line corresponding to a diameter of 40 inches the bursting pressure of the cone will be

$$\frac{7795.62}{20} = 389.78 \text{ } \textit{pounds}$$

per square inch. For a diameter of 16 inches the seam will fail at

$$\frac{7795.62}{8} = 974.45 \text{ } \textit{pounds.}$$

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POWER

Turbines in the United States Navy

The success of the steamships "Lusitania" and "Mauretania," together with several other transatlantic liners equipped with turbines, appears to have created the popular belief that turbines are destined to supplant reciprocating engines in marine practice. Hence, one often hears the United States Navy Department criticized for its apparent slowness in universally adopting the turbine and for equipping many of the later battleships with reciprocating engines.

The fact is, however, that the propulsion of a warship presents an entirely different problem from that of a merchant ship. The problem in the latter case is comparatively simple; fundamentally it is that of carrying a given weight a given distance in a given time, the speed being practically constant. A warship, however, must be able to steam at maximum speed at short notice, yet under ordinary conditions it cruises at half to three-quarters of this speed. Therefore, economy under these conditions is an important factor.

As is well known, the load curve of a steam turbine at constant speed is much flatter than that of a reciprocating engine; consequently, turbines driving electrical generators, as a rule, show fairly good economy at light loads.

In marine practice, however, the rotative speed is a direct function of the load and the economy of a turbine decreases rapidly with a decrease in speed. Hence the turbine-driven warship shows poor economy at cruising speeds.

In the comparative trials of the scout cruisers "Birmingham," "Salem" and "Chester" last year, the "Birmingham," driven by reciprocating engines, showed the best economy for speeds up to twenty and a half knots. Above this speed the "Chester," equipped with Parsons turbines, showed the best results. The "Salem," having Curtis turbines, showed the least economy up to twenty-two and a half knots, above which its steam consumption was between that of the "Birmingham" and the "Chester."

Of the latest ships, the "Delaware" and the "North Dakota" have identical hulls and the same rated power, the former being equipped with reciprocating engines and the latter with Curtis turbines. On the official trials the "Delaware" attained a maximum speed of twenty-one and a half knots with an expenditure of twenty-nine thousand horsepower, whereas the "North Dakota" required thirty-two thousand horsepower when steaming at a maximum speed of twenty-one knots. Since being in commission the "Delaware" has established a record for steam economy while it is claimed that the performance of the "North Dakota" has been disappointing as to economy, this ship ordinarily consuming about one-third more coal than the "Delaware." Furthermore, last spring the "Delaware" steamed from New York upon short notice for a run to Chile and return, a distance of nineteen thousand miles, and without any thorough overhauling of her engines she made the trip to England to represent the United States Navy at the coronation. Upon the return trip the "Delaware" showed a remarkably low steam consumption.

While the "Delaware" has been making this good showing, the "North Dakota" has been lying at the Brooklyn navy yard with one of her turbines out of commission, the shrouding on the blades having corroded and seized upon the stationary parts, thus rendering the ship practically helpless.

Manifestly it would be unfair to condemn the turbine upon the performance of this one ship, but additional opportunity for comparison will be afforded by the new battleship "Utah," which is equipped with Parsons turbines and is soon to be placed in commission. Meanwhile, the Navy Department is to be commended for going slowly.

A Saving of Forty per Cent.

In a steam-power plant the steam for the approximately six-hundred-horsepower-engine output was generated in four two-hundred-horsepower boilers equipped with forced draft operated by a small engine controlled as to speed by the steam pressure. The fuel used was a mixture of Pocahontas and anthracite pea coal.

With clean boilers and proper methods, an evaporation of approximately ten pounds of water per pound of fuel should have been accomplished, while as a matter of fact the ratio of steam made to coal burned was less than six.

When the efficiency of a boiler plant is low it indicates that either the fuel is not properly burned or that the heating surfaces are fouled with soot upon one side or scale upon the other.

With the comparatively smokeless fuel used in this case it is not probable that the fire side of the boilers was so badly coated as to be the sole cause of the poor results.

Inside, the coating of scale necessary for such a poor showing would be so thick that burning of the sheets or tubes would be almost sure to happen.

It would seem that in the handling of the fires either too much air was forced through and around the fuel, chilling the gases far below the fire temperature, or that so little air was furnished that the combustion was incomplete and less than one-half the heat value of the fuel was realized.

It is highly probable that the latter condition obtained. As the amount of air supplied to the furnace was controlled by the steam pressure it is easily seen that when the working pressure was reached the fan engine would slacken its speed and run only fast enough to keep the steam pressure at the point desired.

All of the steam required to operate a six-hundred-horsepower engine can be easily generated by two boilers of two hundred boiler horsepower each, and if four were used for the work it follows that they were underloaded and inefficient. It is ever more probable that this inefficiency was due rather to poor fire conditions than to fouled heating surfaces.

In either case here was an opportunity for an engineer to make a record which would bring profit both to himself and to his employer.

For a six-hundred-horsepower plant, where the evaporation is six pounds of water per pound of fuel and can be raised to ten by intelligent methods, the saving will amount to forty per cent. of the fuel. In one case the fuel cost would be in the vicinity of fourteen thousand dollars per year and in the other eight thousand four hundred dollars.

The difference between these sums represents the amount that might have easily been saved in one plant. It also would have prevented the installation of central-station service, which in the end proved more expensive than the operation under the inefficient conditions which existed when the change was made.

Explosions in England and America

Many things pertaining to boiler operation are treated very differently in England from what they are here. In that country all vessels, pipes, receptacles and valves are classed as boilers if used to hold or convey any fluid, liquid or gaseous, under pressure. They are directly under the supervision of the Board of Trade, which appoints commissions to conduct investigations, fix the responsibility and assess fines in all cases of accident whether insurance is carried or not.

On July 27, at Manchester, Eng., a formal inquiry into the cause of the explosion of a steaming kettle at Salford, last October, was made by a commission appointed by the Board of Trade which is an example of English practice. The boiler, or kettle, as it was commonly called, was used for steaming dyewood. It was supplied with steam the pressure of which was reduced to fourteen pounds per square inch by means of a reducing valve. On the kettle side of this valve a pressure gage had been fixed, but some weeks previous to the explosion this gage was removed while some alterations were in progress, and was not replaced. There were, therefore, no means of knowing whether the reducing valve was doing its work. It was found by the commission that the foreman who removed the valve was to blame for the explosion and he was fined one hundred dollars.

In contrast with this case may be mentioned any one of the cases of explosion of tanks, boilers, pipes or valves in this country, where usually the accommodating coroner, if possible, places the responsibility on some one of the dead, or blandly rejects evidence tending to censure those really guilty.

It is certain that the reckless waste of human life that exists in America cannot go on forever; and it may be both easier and cheaper to meet the question of public safety in the near future than to suffer from the recoil that will surely come if a different attitude is not taken.

Aeroplane Engines

Henry N. Atwood, a New England aviator, has flown from St. Louis to New York. He covered 1365 miles in twelve days, making twenty stops, the actual flying time being twenty-eight hours and five minutes. No individual flight exceeded two hours.

And yet the papers say that his engine is "practically wrecked. It will need overhauling and requires a thorough rest." It has had to have repairs, apparently rod boxes rebabbitted, *en route*. Other newspaper accounts speak of the motor in terms that suggest overtaxation, long sufferance and remarkable endurance.

We realize that failure in the air means more than it does upon the ground, and that something has to be sacrificed to lightness. Skipping which might be tolerated by the chauffeur of an automobile would set an aviator to looking for a good place to land. But, on the other hand, engines in cheap automobiles, subject to all the strain and shock and wet and dirt of a road test, will run for over a thousand consecutive hours. If twenty-eight hours' run in relays of something less than two and a half hours each were anything like the limit for the aëroplane engine in the present state of the art, there would be a great future for flying machines.

Engineering Caliber

An engineer must be able to think quickly when an accident occurs. Machinery may run for months without anything going wrong, but when something does happen the resourceful engineer is the one who will win out.

It is under adverse circumstances that the engineer can make good if he has it in him. If he cannot make good, his employer stands to lose.

In one case as soon as trouble occurs outside help must be called in; but in the other the resourceful engineer and his associates are at hand to immediately remedy the trouble.

Some engineers are born with a mechanical instinct which enables them to see at a glance what is best to be done in case of accident, and instead of reporting to the office that it will take several days to make repairs, and depending upon the office to see that they are made, work such as will permit temporary operation is begun at once, and the loss due to a shutdown is usually limited to a few hours.

If a man has been denied the gift of intuition, he is handicapped to some extent—it takes him longer to see what is best to be done—but if he has even an ordinary amount of mechanical ability, he can make the ordinary repairs. Without such caliber the engine room is no place for him.

It has been held that an engineer will be benefited by making plans as to what he would do in case of accident, but the chances are against any accident happening as the engineer had seen it in his mind's eye, and his plans are liable to go for naught.

A better plan is to read such literature as will make the mind a storehouse for knowledge, and the information thus gained can be brought into use when an accident happens.

The engineer whose advice is asked by his employer is he who has acquired a practical education along steam-engineering lines, and his employer is quick to recognize his ability to decide engineering questions.

Assume that an engine was overloaded far beyond the economical operating point, and it was advisable to install another of equal or greater size. How would you convince your employer that it would be a saving to install a certain type of engine in preference to another, and why one valve gear instead of another?

Steam engineering does not consist merely of the routine duties about the plant; it is the knowing how and why, the ability to determine cause and effect and to put into practice such methods of operation as will produce economical results.

Refrigeration Department

Principles and operation of ice making and refrigerating plant and machinery

Air Cooling and Moisture Precipitation

By F. E. Matthews

Before proceeding to illustrate the method of calculating the amount of refrigeration required to cool a mixture of air and water vapor it may be advisable to define terms.

Air is a mechanical mixture of nitrogen and oxygen in the practically constant proportion of 80 parts of the former to 20 parts of the latter, a very small percentage, about 3 or 4 hundredths of 1 per cent., of which is replaced by carbon dioxide. Into this uniform mechanical mixture water vapor enters in widely varying proportions. When the air contains all the moisture that it can hold it is said to be saturated. The higher the temperature of the air the more water vapor it is capable of absorbing before becoming saturated. At a given temperature saturated air always contains a certain fixed quantity of water vapor.

POUNDS OF AQUEOUS VAPOR IN 1000 CU.FT. OF AIR AT DIFFERENT TEMPERATURES AND PER CENT. SATURATION

At	10%	15%	20%	25%	30%	35%	40%	45%	50%	55%	60%	65%	70%	75%	80%	85%	90%	95%	100%
100° F.	0.282	0.423	0.056	0.705	0.847	0.988	1.130	1 270	1.411	1.552	1.694	1.835	1.976	2.117	2.259	2.400	2.541	2.682	2.823
95° F.	0.244	0.366	0.489	0.621	0.733	0.855	0.978	1.100	1.223	1.345	1.467	1.589	1.712	1.844	1.977	2.089	2.201	2.323	2 446
90° F.	0.211	0.316	0.422	0.527	0.633	0.740	0.847	0.951	1.056	1.161	1.267	1.373	1.479	1.584	1.690	1.795	1.901	2.005	2 110
85° F.	0.182	0.272	0.363	0.454	0.545	0.636	0.727	0.818	0.909	1.000	1.091	1.182	1.273	1.364	1.455	1.546	1.637	1.728	1.819
80° F.	0.156	0.234	0.312	0.390	0.468	0.546	0.624	0.702	0.781	0.859	0.937	1.015	1.093	1.166	1.249	1.327	1.405	1.483	1.562
75° F.	0.133	0.200	0.267	0.334	0.401	0.467	0.534	0.601	0.668	0.735	0.802	0.868	0.935	1.002	1.069	1.135	1.202	1.269	1.336
70° F.	0.114	0.171	0.228	0.285	0.342	0.399	0.456	0.513	0.570	0.627	0.684	0.741	0.798	0.855	0.912	0.964	1.026	1.083	1.140
65° F.	0.096	0.144	0.193	0.241	0.290	0.338	0.387	0.435	0.484	0.532	0.581	0.629	0.678	0.726	0.775	0.823	0.872	0.920	0.968
60° F.	0.082	0.123	0.164	0.205	0.246	0.286	0.327	0.368	0.410	0.451	0.492	0.533	0.574	0.615	0.656	0.697	0.738	0.779	0.820
55° F.	0.069	0.103	0.138	0.172	0.207	0.242	0.277	0.311	0.346	0.380	0.415	0.449	0.484	0.519	0.554	0.588	0.623	0.657	0.692
50° F.	0.058	0.087	0.116	0.145	0.174	0.204	0.232	0.261	0.291	0.320	0.349	0.378	0.407	0.436	0.465	0.494	0.524	0.553	0.582
45° F.	0.048	0.072	0.097	0.121	0.146	0.170	0.195	0.219	0.243	0.267	0.292	0.316	0.341	0.365	0.390	0.414	0.439	0.463	0.487
40° F.	0.040	0.060	0.081	0.101	0.122	0.142	0.162	0.182	0.203	0.223	0.244	0.264	0.284	0.304	0.325	0.345	0.366	0.386	0.407
35° F.	0.033	0.050	0.067	0.084	0.101	0.118	0.135	0.152	0.169	0.185	0.202	0.219	0.236	0.253	0.270	0.287	0.304	0.321	0.338
32° F.	0.030	0.045	0.060	0.075	0.090	0.105	0.120	0.135	0.150	0.165	0.181	0.196	0.211	0.226	0.241	0 251	0.271	0.286	0.301
30° F.	0.027	0.041	0.055	0.068	0.082	0.096	0.110	0.124	0.138	0.151	0.165	0.179	0.193	0.207	0.221	0.234	0.248	0.262	0.276
25° F.	0.022	0.033	0.044	0.055	0.066	0.077	0.088	0.099	0.110	0.121	0.133	0.144	0.155	0.166	0.177	0.188	0.199	0.210	0.221
20° F.	0.017	0.026	0.035	0.043	0.052	0.061	0.070	0.079	0.088	0.096	0.105	0.114	0.123	0.132	0.141	0.149	0.158	0.176	0.176
15° F.	0.014	0.021	0.028	0.035	0.042	0.049	0.056	0.063	0.070	0.077	0.084	0.091	0.098	0.105	0.112	0.119	0.126	0.133	0.140
10° F.	0.011	0.016	0.022	0.027	0.033	0.038	0.044	0.049	0.055	0.060	0.066	0.071	0.077	0.082	0.088	0.093	0.099	0.104	0.110
+5° F.	0.008	0.012	0.017	0.021	0.026	0.030	0.035	0.039	0.043	0.047	0.052	0.056	0.061	0.065	0.069	0.073	0.078	0.082	0.087
0° F.	0.006	0.009	0.013	0.016	0.020	0.023	0.027	0.030	0.034	0.037	0.041	0.044	0.048	0.051	0.055	0.058	0.061	0.064	0.068
—5° F.	0.005	0.007	0.010	0.012	0.015	0.018	0.021	0.023	0.026	0.028	0.031	0.034	0.037	0.039	0.042	0.044	0.047	0.049	0.052
—10° F.	0.004	0.006	0.008	0.010	0.012	0.014	0.016	0.018	0.020	0.022	0.024	0.026	0.028	0.030	0.032	0.034	0.036	0.038	0.040
—15° F.	0.003	0.004	0.006	0.007	0.009	0.010	0.012	0.013	0.015	0.016	0.018	0.019	0.021	0.022	0.024	0.026	0.028	0.029	0.031

It must be remembered, however, that the temperature of the air does not fix the amount of moisture that it contains except in the limiting case of saturation. In general, air is not saturated, and may contain different amounts of water at the same temperature as it varies in degree of saturation; or it may contain different amounts of moisture at the same degree of saturation at different temperatures. Since the amount of water that it is possible for air to hold in suspension increases with increasing temperature, and decreases with decreasing temperature, it is evident that the air may or may not contain less moisture after cooling than before, according to whether the cooling is carried below the temperature at which the air becomes saturated. The accompanying table shows the amount of vapor in pounds per thousand cubic feet of air at different degrees of saturation at different temperatures. At 100 degrees Fahrenheit and 100 per cent. saturation, for example, 1000 cubic feet of saturated air will contain 2.82 pounds of water vapor, while at 75 degrees Fahrenheit, the amount is only 1.33 pounds, or less than one-half that quantity, and at 15 degrees Fahrenheit, it is still further reduced to about one-tenth of what it is at 75 degrees Fahrenheit.

In the general case, air cooling involves cooling not only the mechanical mixture of oxygen and nitrogen, but a large quantity of water vapor as well. If the air contains sufficient moisture so that the cooling brings it to the point of saturation, the heat that must be abstracted from the water vapor will be only that represented by the specific heat of the vapor and the number of degrees cooled through. If it is cooled below the point of saturation, as usually happens in cold-storage practice, not only the specific heat of the vapor but the latent heat of that part of the vapor precipitated as well must be removed. Generally the process is carried still farther and the precipitated moisture is chilled to the freezing point and finally frozen, when not only the specific heat of the liquid but the latent heat of fusion is involved. If the ice is cooled to a lower temperature the specific heat of the ice is also involved.

It is required, for example, to cool 2000 cubic feet of air per minute from 80 degrees Fahrenheit to 36 degrees Fahrenheit. In the following calculations it is assumed that the amount of air to be cooled is 2000 cubic feet before it is cooled, and not, as it might be construed to mean, 2000 cubic feet of cooled air.

For the sake of simplicity the air is assumed to be dry.

Dry air at 80 degrees Fahrenheit weighs 0.0731 pound per cubic foot; 2000 cubic feet would weigh, in pounds, 146.2; the specific heat of air is 0.2377; B.t.u. required to cool 2000 cubic feet 1 degree Fahrenheit, 34.75; B.t.u. required to cool 2000 cubic feet 44 degrees Fahrenheit, 1529.

One ton of refrigeration is sufficient to dispose of heat at the rate of 288,000 B.t.u. per 24 hours, or (dividing this number by 1440, the number of minutes in 24 hours) the equivalent rate per minute is 200 B.t.u.

On this basis, the cooling of 2000 cubic feet of air per minute from 80 degrees Fahrenheit to 36 degrees Fahrenheit would require the expenditure of

$$1529 \div 200 = 7.64 \textit{ tons}$$

of refrigeration.

Had the requirements been for 2000 cubic feet of cooled air, the amount of refrigeration needed would have been 8.36 tons, the difference being accounted for by the difference of weight per cubic foot of air at 80 degrees Fahrenheit and 36 degrees Fahrenheit, respectively.

LETTERS

Answers to E. J. Gale's Questions on Refrigeration

There have been received so many answers to E. J. Gale's questions in the August 8 issue that it would take up too much space to publish each separately. It would also mean unnecessary duplication. A summary of the information they contain has been prepared and is given below:

1. The consensus of opinion is that the discharge from an ammonia compressor should be hot. A Fahrenheit thermometer in the discharge pipe should register from 200 to 260 degrees and the temperature of discharge should increase as the ratio of the absolute discharge pressure to the absolute suction pressure increases. For a condenser pressure of 185 pounds per square inch gage, or 200 pounds absolute, and a suction pressure of 15 pounds gage, or 30 pounds absolute, the discharge temperature should be about 250 degrees Fahrenheit.

2. A hot discharge such as recommended in answer to No. 1 is obtained by returning the ammonia vapor to the compressor in a dry saturated state. This quality of suction vapor can be recognized at the compressor by the frost on the suction line just covering the suction stop-valve body. If the frost leaves the suction pipe in the engine room, the discharge will increase in temperature, only superheated ammonia vapor reaching the compressor.

A cold discharge is obtained when the frost is allowed to travel beyond the suction stop-valve body so that it covers the suction-valve housings and the compressor body. Experience has shown that this mode of operation does not give the maximum capacity per cubic foot of compressor displacement or the lowest horsepower per ton of refrigeration produced.

A hot discharge from superheated vapor can be remedied by increasing the opening of the ammonia expansion or regulating the valves slightly until frost appears on the body of the suction stop valve.

A cold discharge can be brought up to proper temperature by slightly closing the expansion or regulating valves until the frost leaves the compressor. Care must be so exercised in manipulating the expansion valves that the feed of those only is reduced which are allowing too much liquid ammonia to enter the low-pressure side of the plant.

When the return frost just covers the body of the suction stop valve and the discharge is still too cool, the suction valves of the compressor may not be operating properly, only a small quantity of ammonia being discharged from the compressor. In this case the plant will show a falling off in capacity and the trouble should be remedied at the earliest opportunity.

3. The amount that the expansion valve should be opened depends on the conditions under which the cooling coils or other refrigerating apparatus, which they supply with liquid ammonia, are to be operated. For maximum capacity, the expansion valves should be opened far enough so that every square foot of cooling surface can produce the greatest possible cooling effect for the suction pressure carried and the temperature desired in the refrigerator. This condition of operation is obtained when the ammonia vapors flowing through the cooling coils carry enough liquid ammonia with them to the end of these coils to thoroughly wet the cooling surfaces. No liquid, or only slight traces of liquid, should be carried into the suction mains, as otherwise the compressor will frost over, causing a cold discharge and loss in capacity and efficiency.

It is impossible to give any set of fixed rules for regulating expansion valves. The operating engineer must determine the proper opening of each valve under his care used in the plant by carefully noting the action of the system under various conditions of operation.

4. The higher the suction pressure the less horsepower will be required to operate the refrigerating machine. If then the plant is doing the work required at the time and the suction pressure drops, slow down the machine to maintain the pressure that experience has shown to give the best results. If the expansion valves are opened further under the above conditions too much liquid will be carried back to the compressors and the proper operation of the machine interfered with.

5. The compressor jacket water can readily be frozen by opening the expansion valves so much that liquid ammonia is carried back to the machine. It is dangerous to operate most refrigerating machines this way, as the presence of liquid in the compressor may cause a blowing out of the compressor head or other injury to the machine.

6. In a properly designed plant more work can be done with a little ammonia in the system when the condenser pressure is reduced as much as possible so as to obtain the maximum refrigerating effect per pound of ammonia circulated. Experience has shown that in some plants better results are had under these conditions when the condenser pressure is increased. This is because the plant is so constructed that with low condenser pressure, liquid is trapped in the ammonia condensers and does not flow readily into the receiver. These plants are poorly constructed and the fault should be remedied.

7. If an expansion valve is open at all and enough ammonia is in the system, the coil on which the valve is located must frost over to a certain extent as long as the back or suction pressure is low enough so that the ammonia boils below 32 degrees Fahrenheit. To cover the whole coil with frost the valve must be opened far enough so that liquid is carried to within a short distance from the point where the coil enters the suction line. When there is not enough ammonia in the system, part vapor and part liquid will enter the liquid main and only those coils most favorably located will get some liquid; the others will probably get none.

8. This condition of operation indicates too little ammonia in the receiver at times. The lack of ammonia in the receiver may be due to too little ammonia in the system, or else the liquid is periodically trapped in the ammonia condenser. The remedy for the former is to charge more ammonia into the system and for the latter to improve the construction of the ammonia condenser.

9. There is no danger in operating a refrigerating system with a high-suction pressure.

10. To obtain the largest ice output for any plant, the condenser pressure, and the suction pressure as well, should be carried as low as possible. The difference between the two pressures will take care of itself; it depends on the size and the efficiency of the apparatus of the plant and the conditions under which it must be operated.

The above summary was made up from the answers of the following contributors: William Chaddick, Chicago, Ill.; J. P. Colton, Ohio City, O.; C. A. Scott, Wales, Wis.; L. M. Johnson, Glenfield, Penn.; William L. Keil, Philadelphia, Penn; Andrew Blair, Jr., Norborne, Mo.

Flooded System of Refrigeration

In POWER of August 8, Victor Bonn asked for an explanation of the workings of the flooded system of refrigeration. The term "flooded system" as applied to refrigeration may be defined as a method of supplying liquid anhydrous ammonia to the receptacle in which the evaporation of the refrigerant takes place. Briefly, the flooded system of refrigeration is one in which the ammonia is allowed to pass through the freezing coils and then through a trap, which serves as an accumulator, in which the liquid ammonia is separated from the ammonia gas in much the same way that water is removed from steam in a steam trap.

The ammonia gas then passes on from the accumulator to the compressor, and the liquid ammonia is returned by gravity to the freezing coils and passes through the coils again together with the fresh liquid ammonia.

Before proceeding with the principle of operation of the flooded system it is necessary to understand the action of the ammonia in the coils. It is generally understood that the heat abstraction, or the refrigerating effect, is produced by an evaporation of the refrigerant. When any liquid is changed from its fluid condition into a gaseous state a certain amount of heat is necessary to make the change. When water is evaporated in a steam boiler to make steam a certain amount of heat must be supplied even after the water has reached the temperature corresponding to the pressure in the boiler, in order to change it from the liquid to the gaseous state. The extra amount of heat necessary to change the water to steam without raising the temperature of either the water or the steam is known as the latent heat. The same principle that governs the change of water to steam, also governs the change of liquid anhydrous ammonia from the liquid to the gaseous state. When liquid ammonia, in a direct-expansion refrigeration plant, passes through the expansion valve the liquid changes to a gas in its passage through the cooling coils, provided that sufficient heat can be abstracted from the surroundings to supply the necessary amount of heat to make the change. If sufficient heat is supplied the ammonia, which boils at a low temperature, is evaporated, the heat absorbed being abstracted from the surroundings. On the other hand, if so much liquid is supplied that the heat abstracted is not sufficient to evaporate it all, the liquid which remains will be discharged from the refrigerating system and enter the suction line to the compressor. As the liquid ammonia, of course, must not be drawn into the compressor, a trap which contains baffle plates similar to a steam trap may be used in the suction line to catch the liquid ammonia, the gas being allowed to pass on to the compressor. The liquid ammonia in the trap or accumulator may then be allowed to return to the coils and to pass through again, together with the fresh liquid ammonia which is supplied to take the place of the amount vaporized.

This, in brief, is a description of the action of the flooded system of refrigeration. To accomplish the foregoing results, the expansion coils are arranged vertically and the liquid anhydrous ammonia is supplied at the bottom of the coils. Usually a number of expansion coils are used and are connected by headers, the liquid ammonia being supplied to the lower or inlet header. A trap or accumulator is located in the suction line from the top or outlet header of the coils and is placed at a higher level than is the outlet header. As the gas and the entrained liquid ammonia pass into the accumulator the gas is separated from the liquid, the gas passing on to the compressor. The entrapped liquid ammonia in the accumulator is then returned by gravity through a pipe connection provided for this purpose to the inlet header where it again passes through the coils together with the fresh supply. The fresh supply of liquid ammonia from the high-pressure side of the system is not led directly into the inlet header of the cooling coils but is cooled by means of a cooling device while still under the high pressure of the system. After the liquid ammonia at this high pressure has been cooled to the temperature of the low-pressure side of the system the supply enters the accumulator without evaporating and then the supplemental supply and the entrained supply caught in the accumulator flow by gravity to the inlet header and then through the expansion coils where partial evaporation takes place. The pipe line from the accumulator to the inlet header of the expansion coils, as well as the accumulator itself, is so thoroughly insulated that evaporation will not take place until the inlet header of the expansion coils is reached.

By means of the flooded system of refrigeration it is claimed that more work, or a greater refrigerating effect, can be produced with a given amount of evaporating surface than is possible with the direct-expansion system of refrigeration. This is because every square foot of the heat-absorbing surface is efficient as it is in touch with the liquid refrigerant. In the direct-expansion system of refrigeration the amount of efficient heat-absorbing surface depends somewhat on the engineer in charge, while with a properly designed flooded system the regulation of ammonia is automatic and requires no attention. The supply of the refrigerant for the flooded system is controlled by one valve for each accumulator, while with the direct-expansion system one expansion valve is usually supplied for each cooling coil. It is needless to say that one valve feeding a large quantity of liquid is much more easily regulated than many valves handling in all the same quantity of liquid.

In regard to Mr. Bonn's question as to whether the capacity of the plant can be increased without increasing the coal consumption, I would say that such claims have been made for the flooded system, and some plants have been altered under a guarantee to produce a greater output, other conditions remaining the same.

Mr. Bonn also asks if there is any automatic arrangement by which it is possible to maintain an almost constant back pressure. The automatic regulation of the flooded system is probably the arrangement to which he refers, no expansion valves being used and the regulation of the supplemental liquid ammonia being made by a single valve at the accumulator.

T. W. HOLLOWAY.

Scranton, Penn.

Opening an Ammonia Joint

In the June 27 issue, I read with interest D. L. Fagnan's article under the above title. There are some points in the article which I cannot understand. Mr. Fagnan writes: "I rushed for the valves in the engine room to isolate that coil and found one valve partly open." Every modern ammonia system has, or should have, a main liquid valve.

To have performed the pumping-out process properly, the assistant should have closed all liquid valves in the engine room, as there was probably no main liquid valve, and closed the main return valve, or at least the return valve on the coil on which he was working. I have always found it good practice to close all valves on the liquid line, and all return valves unless I was sure that the valves on the coil which was being repaired were tight. The low-pressure gage must be watched while pumping back. It is a good plan to pump down until the gage reads zero or a little below, rap the gage occasionally and see that the hand of the gage works free. Stop the machine when the hand arrives at zero or a little below, and in a short time the hand on the gage will raise. The system cannot be pumped down at the first trial, and it may be taken for granted that it is not empty until the gage will remain at zero. Always break the joint near the valve in the engine room gently, and do not remove the nuts or bolts. After this, break the joint on the coil in which the tee is to be placed. One may meet with some oil mixed with a little ammonia while prying the flanges apart; let the oil drain out before proceeding. If the gage is right a little vapor that may have been in the line will escape through the broken joint into the engine room, and if the valve on the liquid line leaks, the fumes will also escape there. Had the engineer watched his gage and not broken the joint so long as the low-pressure gage indicated pressure, he would have had little trouble. A main liquid valve in Mr. Fagnan's system is needed badly.

WILLIAM L. KEIL.

Philadelphia, Penn.

The development of cheap power on farms where water power is available, says the Geological Survey of Tennessee in its magazine, *The Resources of Tennessee*, will help to solve the problem of "keeping the boy on the farm" and provide a means of running it economically and profitably.

The Christie Air Steam Engine

A 300-horsepower tandem engine, working on the four-stroke cycle, has been built for some months. Actual test figures are awaited to confirm the exorbitant claims made by the inventor.

Boldly claiming to be able to produce 100 per cent. more power with the same boiler and the same fuel consumption, the Christie Engine Company, of Waterloo, Ia., has been formed for the purpose of exploiting the Christie four-stroke cycle air-steam engine. It is the invention of E. J. Christie, of the above town, and an actual machine rated at 300 horsepower has been constructed by the Vilter Manufacturing Company, of Milwaukee, Wis.

The engine, as shown in Figs. 1 and 2, is of the Vilter company's ordinary Corliss type with the exception that it has inwardly opening valves in the cylinder heads to admit air and an arrangement by which the main valves are prevented from opening at every alternate stroke. This is timed to occur on each of the four cylinder ends successively.

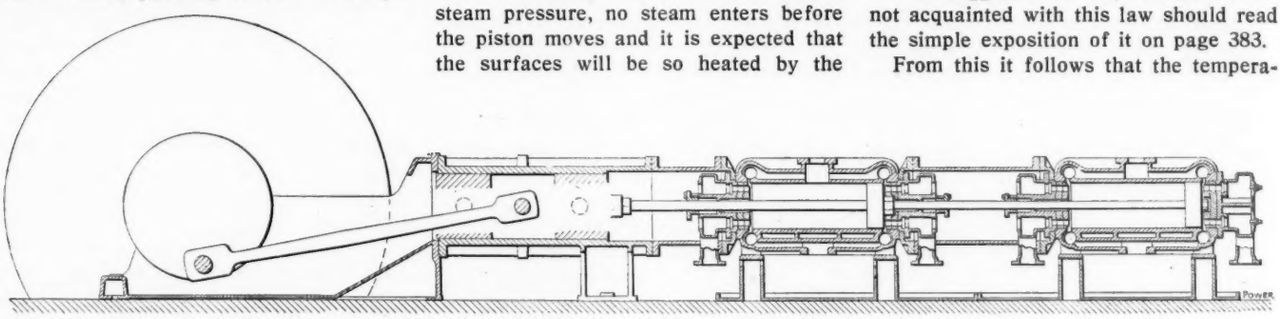


FIG. 1. LONGITUDINAL SECTION THROUGH ENGINE

Each cylinder end works on a four-stroke cycle. With the steam-admission and the exhaust valve closed, the piston moves forward, drawing in a charge of air which is compressed upon the return stroke, the clearance being so proportioned that the pressure attained by compression is that of the entering steam.

At the commencement of the next stroke the steam valve is opened, and as the piston moves away the steam enters, mingling with the air. As the clearance is already filled with air at the steam pressure, no steam enters before the piston moves and it is expected that the surfaces will be so heated by the high temperature of the air due to compression that cylinder condensation will be largely avoided.

It is true that there will be a less active interchange of heat between the hot, dry air and the cylinder walls than with steam; but to just that extent there will be less effective reheating, and it is questionable economy to take expensively produced energy out of the flywheel and reconvert it to heat to warm up cylinder metal. If, on the other hand, initial condensation is avoided, there will be less water to be evaporated during the exhaust stroke at the expense of the heat in the cylinder walls, and hence less reheating to be done.

The leading thought of the inventor is, however, based upon Dalton's laws, which are to the effect that

1. The tension and consequently the quantity of vapor which saturates a given space are the same for the same temperature, whether this space contains a gas or a vacuum.

2. The tension of the mixture of a gas and a vapor is equal to the sum of the tensions which each would possess if it occupied the same space alone.

It is suggested that the reader who is not acquainted with this law should read the simple exposition of it on page 383.

From this it follows that the temperature of the steam in the mixture is determined not by the total pressure but by the pressure which the steam would exert if it occupied the same space alone. Upon this is based the claim that in the Christie engine the steam is highly superheated at cutoff. Also the steam can be expanded down to a pressure below that of the atmosphere without employing a condenser, this being one of the principal claims of the inventor.

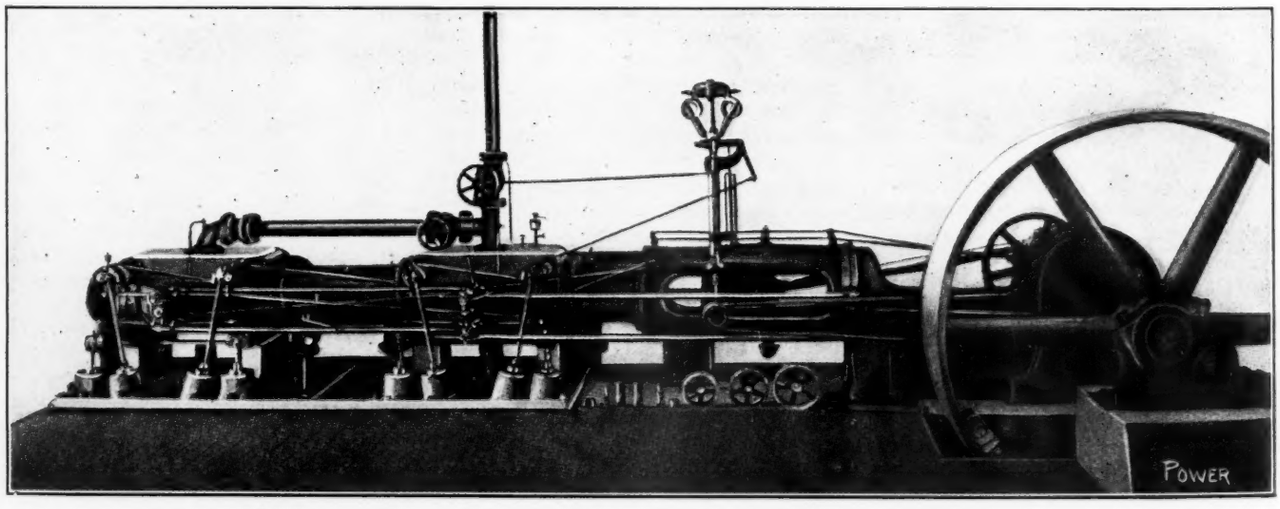


FIG. 2. CHRISTIE AIR-STEAM CORLISS ENGINE RATED AT 300 HORSEPOWER

In the prospectus an example is cited to show that if free air at 100 degrees Fahrenheit be compressed in the engine

up to 165.3 pounds absolute and a temperature of 366 degrees Fahrenheit, and if steam entering the cylinder at the latter pressure and temperature be cut off at such a point as to secure a terminal steam pressure in the mixture of 4.09 pounds absolute and a temperature of 154 degrees Fahrenheit, the air pressure in the exhaust will be 16.12 pounds absolute and that of the mixture will be 20.21 pounds.

Taking these conditions and treating the problem from a thermodynamic viewpoint, at the beginning of compression the air is saturated with vapor. Calculations show that of the total pressure of 165.3 pounds at the end of compression, 154.8 pounds is due to the air and 10.5 pounds to the vapor which is superheated about 170 degrees.

Conforming to the conditions as stated—that is, a terminal air pressure of 16.12 pounds absolute and a temperature of 154 degrees Fahrenheit—the ratio of expansion of the air would have to be 1283.

The engine has two 14-inch cylinders and a 36-inch stroke. Therefore, the piston displacement in each cylinder is 3.21 cubic feet per stroke. Since the ratio of expansion is 12.83, the volume of the clearance must be

$$V = \frac{3.21}{12.82 - 1} = 0.2814 \textit{ cubic feet}$$

Hence the total volume of each cylinder is

$$3.21 + 0.2714 = 3.4814 \textit{ cubic feet}$$

At the terminal steam pressure of 4.09 pounds the density of the steam is 0.01131 pound per cubic foot. Therefore, the theoretical steam consumption per stroke per cylinder would be

$$3.4814 \times 0.01131 = 0.03938 \textit{ pound}$$

and at 150 revolutions per minute, the steam consumption per hour for the two cylinders would be

$$0.03938 \times 150 \times 60 \times 2 = 709 \textit{ pounds per hour}$$

The inventor claims that the engine is able to develop 300 horsepower when running at 150 revolutions per minute. If this were so, it would mean a steam consumption of

$$\frac{709}{300} = 2.36 \textit{ pounds}$$

per horsepower-hour, the absurdity of which is at once apparent.

Again, referring to Fig. 3, if the cycle operates as claimed, the expansion of the air from steam admission to cutoff would be isothermal as represented by AH and from H to B the expansion would be adiabatic. Since the volume of the air at the end of adiabatic expansion is 3.4814 cubic feet and the temperature is 154 degrees, at the beginning of expansion with a temperature of 366 degrees the volume would be 1.686 cubic feet. Hence, the cutoff would have to be at

$$\frac{1.686 \quad 0.2714}{3.21} = 45 \textit{ per cent.}$$

of the stroke and the air pressure determined from the formula

$$\frac{t_1}{t_0} = \left(\frac{P_1}{P_0}\right)^{0.41}$$

would be 45.06 pounds. At the beginning of the isothermal expansion the air conditions are as follows:

Temperature = 366 degrees Fahrenheit.
Pressure = 154.8 pounds absolute.
Volume = 0.02714 cubic foot.

At the end of this expansion and to suit the adiabatic expansion, which starts at cutoff, the air conditions should be:

Temperature = 366 degrees Fahrenheit.
Pressure = 45.05 pounds absolute.
Volume = 1.686 cubic feet.

For isothermal expansion of air

$$p\,v = p_1\,v_1$$

Hence the expression

$$154.8 \times 0.2714 = 42.04$$

should equal

$$45.05 \times 1.686 = 76$$

Since there is such a wide difference in these values, it follows that the engine cannot operate as claimed without destroying the fundamental principle upon which the engine is supposed to operate.

Up to the point of cutoff the total pressure is supposed to remain constant at 165.3 pounds absolute, which is the sum of the air and steam pressures; as the air pressure drops the steam pressure rises. Therefore, at admission the effective steam pressure would be negligible, but this would increase as along the curve EF to a pressure of

$$165.3 - 45 = 120.3 \textit{ pounds}$$

at cutoff. It then expands to 4.05 pounds absolute. It is evident therefore that, although the boiler pressure is 165.3 pounds absolute, the mean effective pressure of the steam will be very small.

Of course, the air cannot be looked upon as performing any useful work; it is merely a medium filling the clearance spaces, being supposed to lessen condensation and permit an expansion of the steam below atmosphere; in fact, it is a question whether the air does not introduce more losses than it prevents.

The engine has now been built for some months, affording ample opportunity for testing its merits and substantiating the claims of its inventor. Test figures giving the water rate of the engine have been requested on several occasions, but these have never been forthcoming, although the inventor continues to sell stock. Data on the steam consumption will do more than a volume of theory to establish the claims for the engine or condemn it. The actual test figures will be awaited with interest.

Dalton's Laws

Suppose the vessel in the accompanying sketch to have a capacity of three cubic feet and to be divided as in Fig. 1 into a chamber containing one cubic foot of steam at 150 pounds, absolute, pressure. Now suppose the partition to disappear. The steam would expand to three times its volume, and, since no work is done, its temperature would remain constant. Under these conditions its pressure would vary inversely as its volume and would be

$$\frac{150}{3} = 50 \textit{ pounds}$$

The pressure is due to the bombardment of the containing surfaces by the molecules of the steam. The temperature of the steam depends upon the velocity of its molecules. If the velocity (that is, temperature) is kept constant and the volume trebled, the molecules, which were traveling lengthwise of the cylinder, will have three times as far to go between impacts, and at the same velocity can hit only one-third as often. Those which have a crosswise direction have three times as much area to cover. The number of impacts per unit of surface therefore becomes one-third for the increased volume, with a proportionate reduction of pressure.

Now imagine a similar cylinder, Fig. 2, where the smaller compartment is vacuous and the larger full of air com-

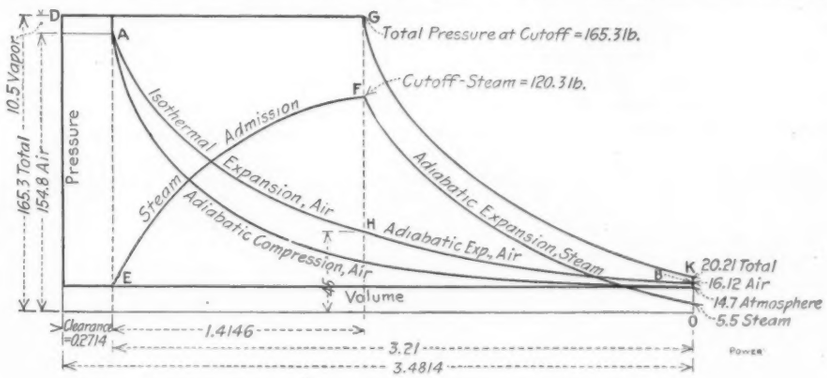


FIG. 3. AIR-STEAM PRESSURE DIAGRAM

pressed to 150 pounds absolute. If the partition were to disappear the two cubic feet of air would expand to three cubic feet, the temperature would remain constant and the air would be at a pressure of

$$\frac{150 \times 2}{3} = 100 \textit{ pounds}$$

Again imagine the cylinder, as in Fig. 3, to have steam on one side of the partition and air on the other, both at the same temperature and pressure. If now the partition should disappear, what would happen? The steam would not stay in one end of the cylinder and the air in the other, but the molecules of each would be diffused throughout the whole vessel.

The pressure due to the steam would fall just as it did when the air was not there, for the impact of its molecules would be divided over all the cylinder wall as before. The pressure of the air would fall in the same way. But the walls are subjected to the bombardment of both the steam and the air molecules so that the pressure upon them is the *sum* of the air and the steam pressures; and since that of the air is 100 and that of the steam 50 the actual pressure is

$$100 + 50 \textit{ pounds per square inch}$$

as before.

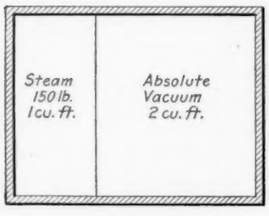


FIG. 1.

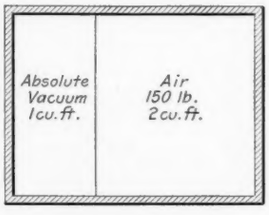


FIG. 2

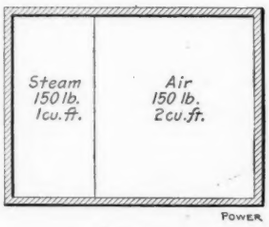


FIG. 3

With the above in mind it is easy to understand the laws of Dalton, which may be expressed as follows:

First Law: The pressure and consequently the quantity of vapor which saturates a given space are the same for the same temperature whether this space also contains a gas or is vacuous.

This simply means that if a pound of steam molecules is put into a certain space and given the velocity corresponding to a certain temperature it will exert a certain pressure. If there are other molecules flying about in the same space they will exert their pressure too. If an attempt is made to put more steam into the same space without changing its temperature, the pressure will go up, because there will be more molecules to impact per second at the same velocity.

Second Law: The pressure of a mixture of a gas and a vapor is equal to the sum of the pressures which each would possess if it occupied the same space alone.

If in a condenser, for instance, there is a vacuum of 26 inches (say an absolute pressure of 4 inches of mercury) and the temperature of the condensate is 110 degrees, the condenser must be full of vapor of 110 degrees temperature, which the steam table tells us corresponds to a pressure of 2.589 inches of mercury. There must then be air enough present to exert a pressure of

$$4 - 2.589 = 1.411 \textit{ inches}$$

Novel Method of Overcoming Peak Load Troubles

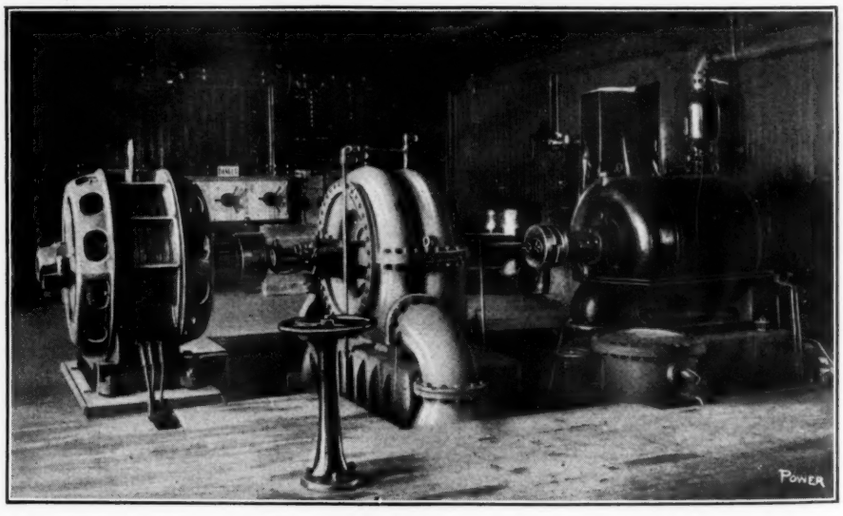
Users of electric power are sometimes compelled to buy on a "peak-load" basis; that is, not exceeding a fixed current consumption at any time during the year but paying constantly for that fixed maximum regardless of how much current is used. On this basis it is evident that continually running close to the specified limit or *vice versa*, keeping the current consumption low at the time of greatest load is advantageous. The former procedure is impossible in many manufacturing processes, as it is in lighting and street-railway work; the latter is usually attempted by the use of storage batteries.

Those who have studied this peak-load problem will be interested in a novel soultion which has been quite successful in the municipal lighting and waterworks plant of Lachine, Canada. In this instance, the low cost and satisfactory operation of a 400-horsepower steam-turbine unit are utilized in effecting the economy. In the Lachine plant, a 14-inch, two-stage, double-suction turbine pump with a capacity of 6,000,000 gallons per 24-hour day is utilized to provide water supply and fire protection for the city, the water pressure being ordinarily 80 pounds per square inch and for fire 120 pounds per square inch. The lower pressure is obtained by closing one valve and operating only one stage of the pump. The speed remains the same for both services.

Current is purchased by the year on a basis which is all right for lighting the city and for pumping at all times of the year except the three winter months during which the lighting load is greatest. It was figured that operating the pump by steam for four or five hours a day during that period could be made less expensive than buying sufficient electric current to operate entirely by electricity. As further advantages of an auxiliary steam installation, the insurance rate on the pumping station could be reduced and the city could be better lighted during fall evenings without incurring unreasonable additional expense.

It was therefore decided to arrange the pump for operation by a 400-horsepower induction motor the greater part of the time but by a steam turbine during the peaks in the lighting load in winter and in case of accident in the electric line.

The installation, furnished by the John McDougall Caledonian Iron Works, Montreal, is shown in the accompanying photograph. The pump, of Worthington make, runs at 1200 revolutions per minute and is direct connected to an Allis-Chalmers-Bullock motor on one side and a 400-horsepower four-stage Kerr turbine on the other. Either driving unit can be thrown into or out of use instantly by means of clutches on the shaft.



MOTOR-TURBINE-PUMP INSTALLATION AT LACHINE, CAN.

High Duty Engine at Providence

About a year ago the old Nagle engine at the Hope station of the Providence water works was replaced by a Worthington high-duty, triple-expansion pumping engine of 10,000,000 gallons capacity.

The new engine has two high-pressure steam cylinders, each 16 inches in diameter, two intermediate cylinders 25 inches in diameter and two low-pressure steam cylinders of 46 inches diameter. The two double-acting water cylinders are each 24 inches in diameter and the common stroke is 24 inches. The engine is of the self-contained type with three steam cylinders for each side arranged

FIG. 1. HOPE STATION OF PROVIDENCE WATER WORKS

tandem, each set of steam cylinders being directly connected to one double-acting water plunger. Each of the steam cylinders has two admission valves on top and two exhaust valves on the bottom, all of a modified Corliss type. The former are provided with a nonrelease cutoff gear operated by a secondary four-arm crank which is fulcrumed on the wristplate of the main valve gear but receives its motion from its own side of the engine, while the wristplate of the main valve gear is moved from the crosshead of the opposite side in the ordinary manner of the duplex-valve gear.

The high-duty attachment is of standard design, each side of the engine having two oscillating cylinders containing plungers attached to the main crosshead. These cylinders are under constant pressure from the discharge main and this is increased to the required amount by an interposed differential accumulator. The effect of the plungers is to resist the advance of the piston rod at the beginning of the stroke and assist it at the end. By thus alternately absorbing and exerting energy, due to the different angles at which the force is applied in relation to the motion of the water plungers, these compensating cylinders perform the function of a flywheel, and, what is most important, the power they exert increases in almost exact proportion to the decrease in the power of the expanding steam.

To eliminate the possibility of the engine short-stroking under a variable load, a stroke-adjusting device of recent development is attached which automatically maintains a practically constant stroke. This device is so arranged as to change the back pressure of the accumulator whereby the compensating cylinder load is correspondingly increased or decreased as may be required for the lengthening or shortening of the stroke of the engine.

The pump ends are of the sectional plunger and ring type with the suction and discharge chambers solidly bolted to the pump barrels. The latter are provided with horizontal valve decks containing multiple water valves of small diameter but of sufficient area to insure quiet operation. The condenser is of the surface type through which passes all of the water pumped by the engine.

The air pump is independent and of the crank and flywheel type. It is located in the basement of the station and receives steam from the main standpipe and exhausts into an auxiliary feed-water heater located in the boiler room through which the feed water, after being passed through the main exhaust heater, is fed to the boilers by an independent feed pump.

The results obtained at the official trials of this engine are given in the accompanying table, taken from the city engineer's report.

Test No	1	2	3
Duration, hours	6	3	4
Steam pressure, pounds gage	151.4	151.2	151.9
Vacuum, inches of mercury	27.45	27.67	27.75
Barometer, inches of mercury	29.94	30.02	30.00
Discharge pressure, pounds gage	73.14	79.94	73.64
Suction pressure, pounds gage	5.13	5.70	6.37
Total pressure pumped against, pounds	67.64	73.70	66.72
Average stroke of engine, inches	24.75	24.68	24.77
Revolutions per minute	39.38	27.22	25.74
Piston speed, feet per minute	162.4	112	106.3
Million gallons of water pumped	2.713	0.935	1.183
Capacity, rate per 24 hours, million gallons	10.85	7.48	7.10
Total steam used, pounds	25,422	9,832	11,475
Moisture in steam, per cent	1.71	2.38	2.38
Total dry steam used, pounds	24,987	9,598	11,202
Duty per 1000 pounds steam used, million foot-pounds	138.6	135	132.6
Duty per 1000 pounds dry steam, million foot-pounds	141	138.3	135.8
Water horsepower	296.7	223.5	192
Dry steam per water horsepower-hour, pounds	14.04	14.31	14.59

Additional credit for the heat returned to the boilers as provided for in the contract, is not included in the foregoing figures. With this allowance, however, the engine is credited with a duty of 149.5 million foot-pounds per 1000 pounds of dry steam when operating under full load and with a duty of 142 million foot-pounds per 1000 pounds of dry steam when delivering 7,000,000 gallons for 24 hours, exceeding the guarantee 10.5 and 18 per cent. respectively.

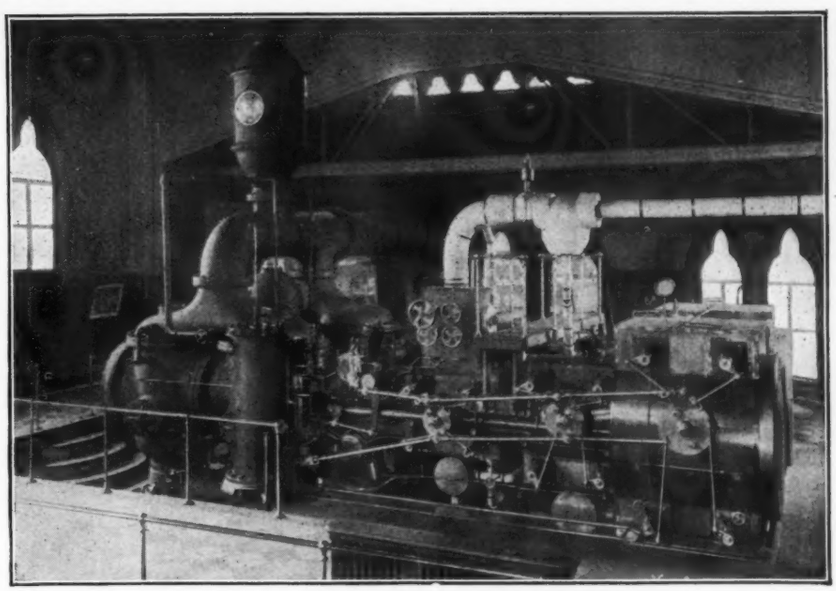


FIG. 2. WORTHINGTON TRIPLE-EXPANSION 10,000,000-GALLON PUMPING ENGINE

To demonstrate how the engine would operate under the various conditions met with in actual service, additional trials were made by pumping into the open service with the suction taken from the Hope reservoir and again with the suction taken from the street mains, also by pumping into the closed service.

New Power House Equipment

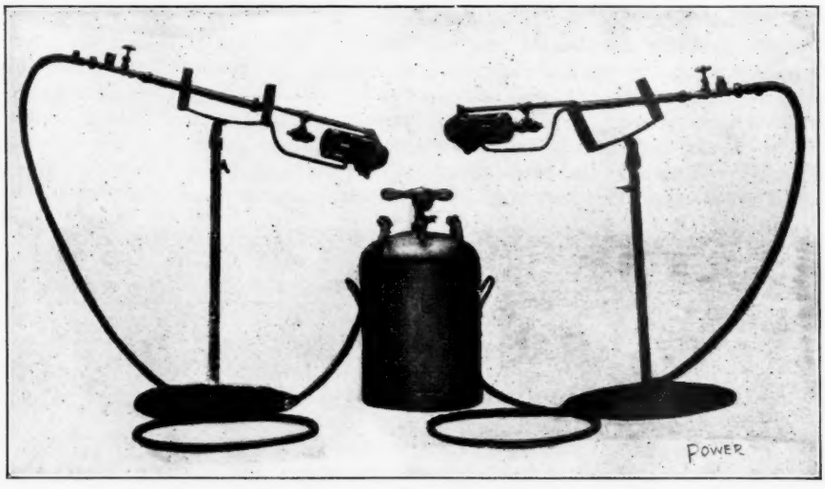
What the inventor and the manufacturer are doing to save time and money in the engine room and power house. Engine room news

Portable Oil Burning Outfit

The accompanying illustration shows a Hauck portable oil-burning outfit, adapted for all kinds of repair work, brazing and preheating in connection with electric or oxyacetylene welding, as well as tempering, annealing, melting, metals and similar operations wherever heat is required.

The outfit consists of a seamless tank, equipped with a hand air pump, two sets of Hauck burners and hose attached to the tank and two adjustable stands to hold the burners in a proper position.

The burners use kerosene oil as a fuel and give very powerful, clean flames, which are always under the control of the operator. This outfit is manufactured by the Hauck Manufacturing Company, Richards street and Hamilton avenue, Brooklyn, N. Y.



PORTABLE OIL-BURNING OUTFIT

Blake-Knowles Slush Pump

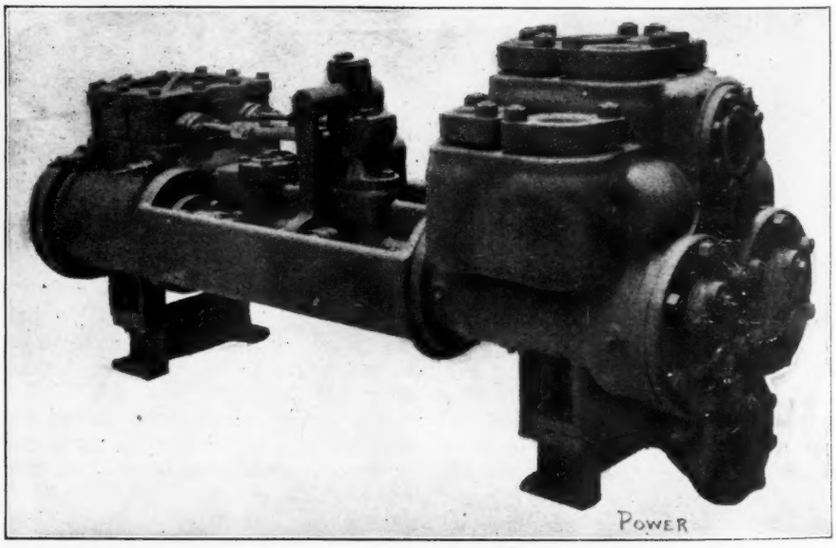
The Blake & Knowles Steam Pump Works, East Cambridge, Mass., have designed a pump especially adapted for oil- and gas-well work in pumping "slush" water. It is shown in the accompanying illustration.

This pump is said to be fully 100 per cent. heavier than the pumps heretofore used for this kind of service. The pump cylinders are fitted with heavy and very hard cast-iron removable liners, to resist the scouring action of the slush coming from the well. The liners are flanged and are held in place with studs and nuts. The jointing material consists of an endless round rubber ring forced into a triangular space in a corner of the flange of the removable cylinder. This permits of replacing liners without delay, and forms a tight joint. The pump pistons are made unusually heavy, with specially deep packing spaces. The piston rods are divided so that when necessary to renew the pump rod which works in the water cylinder, the steam rod, which is not subject to the excessive wear, need not be interfered with or discarded.

The water-valve seats are screwed in place and fitted with a special design of valve bolt having a screw-driver slot so as to do away with the use of wrenches. The valves in the pump cylinder are of a special vulcanized rubber composition, made extra thick and reinforced with a metal casing. The rubber is of proper firmness to prevent particles of sand and rock getting in between the valve and seat and keeping the valve from seating properly by imbedding themselves permanently into the face of the valve.

The pump has been so designed as to permit quickly removing and replacing the parts that are likely to wear. Pumps for this class of work are subject to excessive wear.

Another desirable feature is the arrangement of the stuffing-box glands on the water ends. The severe service will occasionally cause breakage of the glands, due to the extreme difficulty in packing against this slush, and when one breaks it would ordinarily mean dismantling the pump to remove the old gland and slip a new one over the rod. To overcome this, the glands on the pump end are made in halves, strongly bolted together so that a broken gland can be quickly and easily replaced. The



BLAKE-KNOWLES SLUSH PUMP

stuffing boxes on the pump end of the piston rod are made twice as deep as those ordinarily used with such pumps to facilitate packing the rods without screwing up the glands excessively hard.

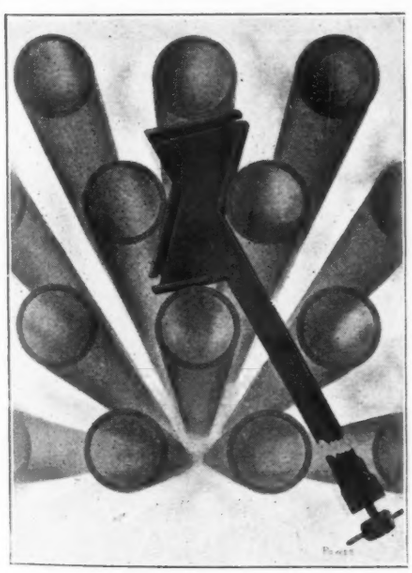
In the fittings that come with the pump special jacks are provided for removing the pump pistons from the piston rods without having to withdraw the rod from the pump. Such a jack is also provided for removing the pump-cylinder liners, which can be quickly taken out and replaced without removing the piston rod.

The working parts of the valve gear and stuffingbox glands are fitted with large lubricating oil wells into which can be placed wicking for carrying the oil. This permits proper lubrication and at the same time does not require frequent attention.

Heely Boiler Tube Spreader Tool

In the accompanying illustration is shown an improved tool used for spreading the tubes of Babcock & Wilcox boilers. The original tool was described in the February 21 issue of POWER.

The improved tool is provided with a wide face on the adjustable spreader head, which prevents any possible injury to the tubes. With the tubes spread as shown it is a simple matter to remove an old baffle wall and replace it



HEELY BOILER-TUBE SPREADER TOOL

with new baffle brick without injury to either tubes or bricks. In this design of tool the extension end has been eliminated.

No improvement has been made in the second tool of the set, which is used for replacing the baffle brick in position and consists of a gripping member, adjustable from the end by a threaded rod.

These tools are manufactured by the Heely Tube Spreader Company, 346 Broadway, New York City.

Type "A" Double Expansion Joint

The Central Station Steam Company, of Detroit, Mich., has recently placed on the market a new and improved type of diaphragm expansion joint. The accompanying interior view of a double joint shows the construction and method of operation.

The double joint consists of two annular diaphragms of heavy cold-rolled and annealed copper, clamped at their outer edges between a cast-iron inner ring and two cast-iron outer rings and, having their inner edges spun through and around the inner edges of the cast-

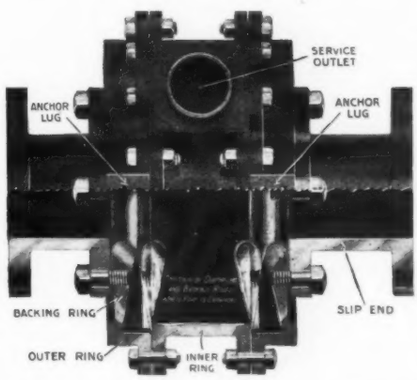


FIG. 1. INTERIOR VIEW OF DOUBLE-EXPANSION JOINT

iron backing rings. The copper is brought up far enough on the outside face of the backing ring to permit clamping it securely between the backing ring and the slip end, which is a short, flanged, cast-iron nipple. The outer ring, inside of and concentric with which the backing ring is located, is recessed to a depth considerably in excess of the thickness of the backing ring.

In operation the joint is placed in the pipe line with the slip ends, and the backing rings drawn out to their farthest limit. The inner and outer rings are rigidly anchored by cast-iron lugs into the concrete or brick box built around the joint; service pipes to the buildings on each side of the street are connected to the service outlets. As the pipe expands the slip ends and backing rings move toward the middle of the joint and the copper diaphragm, which touches the backing ring only at the inner edge when the pipe is contracted, is now drawn close over the backing ring which reinforces it and carries the pressure of the steam.

Fig. 1 shows an interior view of the expansion joint. Very few parts enter into the construction of this device, a double joint capable of taking up the expansion in 100 feet of pipe line having only nine parts exclusive of bolts and anchors, and the maximum bending of the copper at any point, due to change of position, cannot exceed 10 degrees.

A large factor of safety has been

FIG. 2. EXTERIOR VIEW OF DOUBLE-EXPANSION JOINT

provided in the length of traverse, the joints being designed to permit 1¼ inches in 50 feet.

The backing ring, which forms the reinforcement for the copper diaphragm, is one solid heavy iron casting, presenting a smooth unbroken surface to the copper.

The single joint is similar in construction to the double joint except that it has but one diaphragm and is designed to take up the expansion in only 50 feet of line.

New Test for Mineral Oils

Alexander E. Outerbridge, Jr., at the recent meeting of the American Society for Testing Materials, gave a description of a method which he has discovered for detecting the presence of mineral and resin oils when mixed with linseed and other animal and vegetable oils. He found that the greenish bloom or fluorescence of mineral oil and the blue bloom of resin oil can be enormously intensified or magnified perhaps a thousand times, so that samples which have been debloomed and show no fluorescence in sunlight or electric light give a very visible indication.

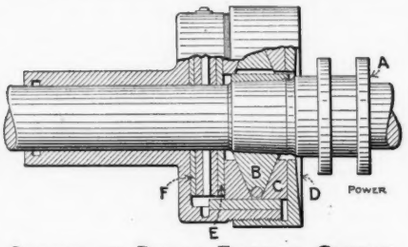
Light from an inclosed arc lamp is used for viewing the specimens, and, although the author does not describe his apparatus, the inference is that the effect is produced by the action of the ultra-violet rays. Samples containing a very small percentage of mineral or resin oil are said to show the bloom or fluorescence very markedly when viewed with this light against a black background, and by comparison with a set of samples with a known percentage of the oil a quantitative determination can be made with fair precision.

Reilly Friction Clutch

The illustration shows a sectional view of a Reilly clutch which is made in cut-off-coupling style. The standard clutch is made solid or split and can be put on the shaft without taking it down.

This clutch is perfectly smooth on the outside, thereby eliminating any chance of accidents. On the larger sizes, a large space inside for oil has been provided which allows the clutch to run in oil, thereby increasing the life of the clutch. The clutch body runs on a hardened and ground bushing which protects the shaft from wear and makes an excellent bearing for the clutch body. On the larger size the shaft body is protected from wear by an additional bushing.

Referring to the figure the sliding wedge *A* enters the segmental cone *B* which forces the thrust collar and driving members *C*, therefore increasing the length between the adjusting head *D* and the bottom of the clutch body, transferring the thrust on the loose member through the collar *E* which in turn transfers the drive onto the driving member *F*. This gives four driving surfaces which act almost like a multiple-disk clutch. It is manufactured by A. S. Baldwin & Co., Sharon, Penn.

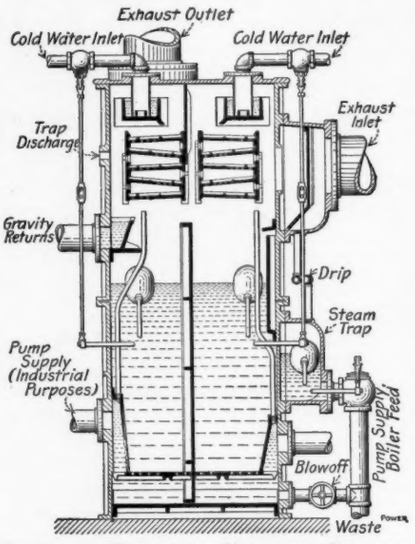


SECTION OF REILLY FRICTION CLUTCH

Cochrane Double Feed Heater

A special boiler feed-water heater for heating two separate water supplies is illustrated herewith. The heater is manufactured by the Harrison Safety Boiler Works, Seventeenth and Clearfield streets, Philadelphia, Penn. The heater is, in general, of the Cochrane open type, receiving the exhaust steam through an attached oil separator. The water supplies, however, are received in two separate distributing boxes which deliver to separate heating trays, from which the water drops into separate storage chambers. The idea is that in many plants there are two different water supplies which are used for two different purposes, as where the returns or condensate from surface condensers is used for washing or dyeing, while city or well water is used in the boilers.

This arrangement represents a considerable saving on cost of heaters, and of valves, piping, etc., while taking up much less room than one or more heaters would occupy. The arrangement is to be used with the several modifications of the Cochrane heater.



SECTIONAL VIEW OF A COCHRANE DOUBLE-FEED HEATER

Vulcan Soot Cleaner as Applied to Economizers and Manning Boilers

The application of the Vulcan tube cleaner to economizers and Manning boilers is illustrated in Figs. 1 and 2.

The accumulation of soot and fine particles of coke in the soot chamber of economizers between the lower headers where the tube scrapers do not reach causes extra labor and loss of heat. This objectionable feature is overcome by the application of the steam tube cleaner, illustrated in Fig. 1, which shows a sectional elevation and cross-sectional view. When steam is turned on the accumulated soot is blown toward the right, as indicated by the direction in which the nozzles incline.

Fig. 2 shows the blower as applied to a Manning boiler. It is attached at the top and is operated from the floor below (the smoke-bonnet lids remaining closed), without interfering in the least with the firing of the boiler.

The blower arm *F* is fitted with a series of nozzles which throw concentrated blasts of dry steam into each tube. This blower arm is turned by means of beveled gears and it revolves on a pivot bearing placed at the center of the top tube sheet and blows the tubes in rows on radial lines.

When about to operate, first the drain valve *A* is opened to blow the condensation from the steam line; the wheel *B*, which opens the main valve to admit the steam through the nozzles, is then opened and the third wheel *D* is slowly turned by pulling the chain *C* gently about 4 inches at a time, so as to allow the steam to blow a few moments through each radial row of tubes. This device is manufactured by the Vulcan Soot Cleaner Company, Du Bois, Penn.

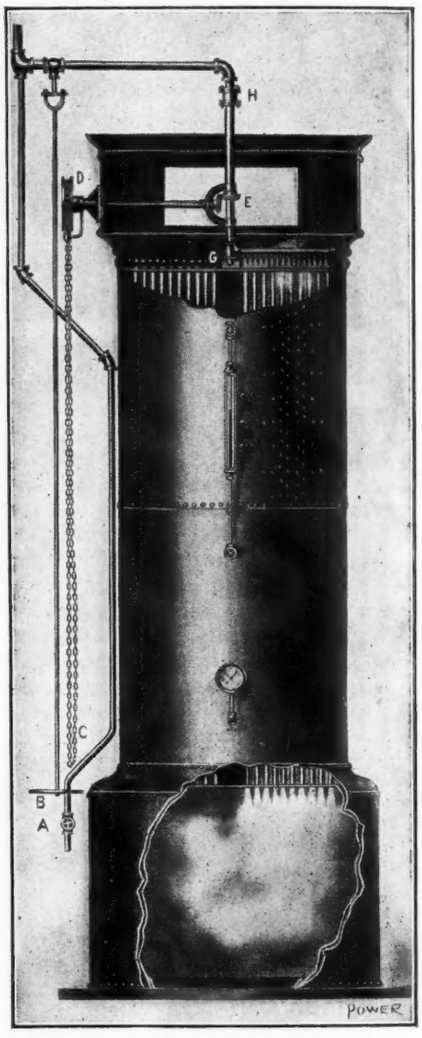


FIG. 2. CLEANER APPLIED TO MANNING BOILER

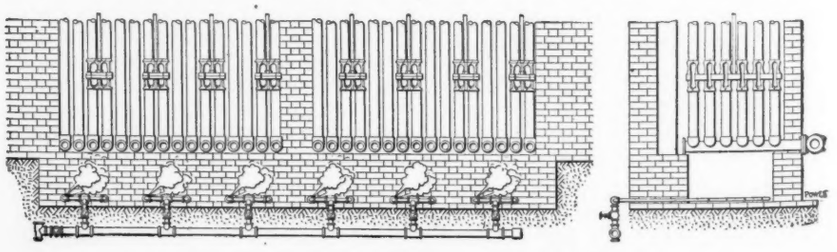


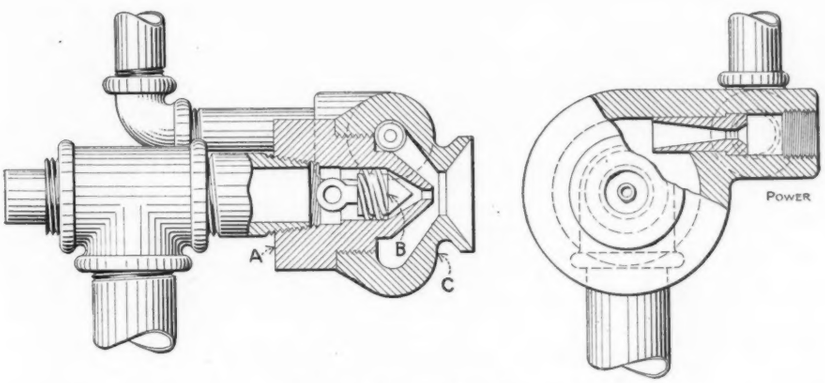
FIG. 1. CLEANER APPLIED TO AN ECONOMIZER

Stilz Fuel Oil Burner

The Stilz burner is shown in section in the accompanying illustration.

The nozzle comprises a central fitting *A* through which the oil under pressure is forced. A spiral *B* located within *A*, near its delivery orifice, causes the oil to acquire a high rate of rotation upon passing through the orifice, so that the actual direction of motion of the oil upon leaving the orifice spreads it out as over the surface of a cone.

Surrounding *A* is an outer casing *C* having a delivery orifice larger than the oil-delivery orifice but sufficiently restricted to permit the steam, which is used as an auxiliary atomizing means, to catch hold of the film of oil and swing it into a much larger angle of divergence and at the same time disintegrate it into a high degree of atomization. As all the high velocities are rotative, the oil is spread over a large sectional area and moves forward into the furnace at a rate about equal to that at which the air would be drawn into a suitably constructed furnace, thus affording a uniform mixture for combustion.



STILZ FUEL-OIL BURNER

The nozzle is located in front of a hole arranged in a wall of firebrick. All the air entering the furnace must pass through this hole, likewise all the atomized oil. Both are moving at about the same rectilinear velocity and the mixture completely fills the hole in the firebrick, as it enters the furnace.

As soon as the brick wall of the furnace becomes well heated at its inner side, the supply of steam to the nozzle may be discontinued, provided the pressure behind the oil is sufficiently great to spread the oil spray so as to just clear this hot ring of firebrick. In this case the oil spray and the current of air, coming together while in contact with this igniting surface, continues the combustion. The supply of air is regulated by suitable air louvers on the furnace front plates.

This burner is manufactured by H. B. Stilz, Vallejo, Cal.

Hinge Edge Conveyer Belt

A conveyer belt generally gives out at the point where it bends between the horizontal and side pulleys, at which point side crimp will occur, caused by the angular bending and intensified by the sag between the carriers under heavy load. This action is illustrated in Fig. 1.

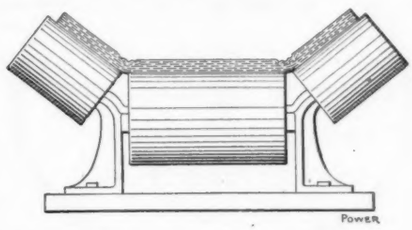


FIG. 1. WHERE BELT CRIMPS ON ROLLERS

The prime requisite for any conveyer belt formed of plies vulcanized or frictioned together is the strength, vitality and longevity of the friction bond which binds and holds the plies together. The Hinge Edge belt represents in operation practically three flat belts.

The tensile strength in conveyer belts is so much in excess of the actual requirement as to make this question of tensile strength a negligible quantity. The necessity for rigidity in the center and sides and for maintaining maximum flexibility at the bending or troughing points makes different weights of duck practically a necessity in order to obtain the best results. This method of construction is illustrated in Fig. 2, which shows that at the bending points of the belt but three plies of canvas are inserted in the rubber which surrounds them. At the center and outer edges there are seven plies of canvas.

To withstand the necessary wear and tear of abrasive material a protective cover or pad is used, and this is compounded to conform to the character of the work to be done.

Hinge Edge belts are made in three grades of cover. Grade "A," to conform to the requirements of heavy ore and gold-dredge work; Grade "B," for the ordinary class of mining, milling, cyaniding and concentration work, and Grade "C," for handling coal, sand, gravel and similar work.

This belt conveyer is manufactured by John J. Ridgway, 207 Fulton street, New York City.

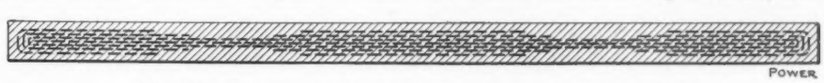
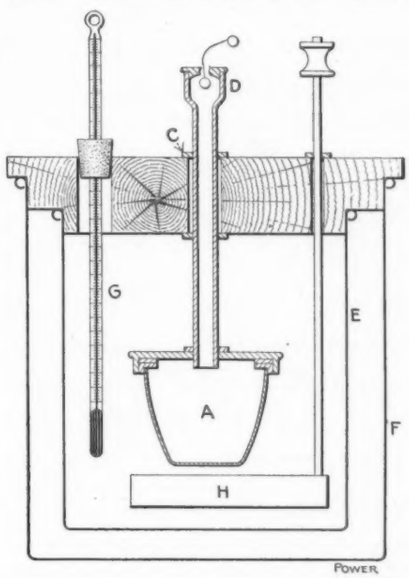


FIG. 2. CONSTRUCTION OF HINGE EDGE CONVEYER BELT

Wild's Calorimeter

This device, designed for estimating the heating value of solid fuel, has recently been put on the market by the Precision Instrument Company, Detroit.

It is of the sodium-peroxide type in which a mixture of the fuel and sodium peroxide is fired in a combustion chamber that is immersed in water.



CROSS-SECTION OF WILD'S CALORIMETER

The apparatus shown in the illustration consists of a combustion chamber *A* suspended from the cover *B* by a conduit *C* which is furnished with a valve *D; E* is a copper water vessel and *F* is an outside vessel, heavily nickeled, forming an air jacket which effectually prevents radiation and absorption, thus rendering the rise in temperature of the water a guide to the heat given off by the fuel. A Fahrenheit thermometer is shown at *G*, the scale being divided into tenths and easily read in twentieths; *H* is an agitating paddle or stirrer.

Each instrument is simple and gives direct readings of calorific value and evaporative power of any kind of coal.

Moments with the Ad. Editor

A department for subscribers edited by the advertising service department of Power

A manufacturer with a new product came into the office a few days ago and wanted information regarding advertising.

We gave it to him and showed him how our papers reach and are read by the men in responsible charge of plants in their various lines, how advertising in the Hill papers would be a good investment for him, provided his product had merit.

We explained to him our policy of inserting advertisements from reliable concerns only, and stated that inasmuch as he was not known to us, we would be obliged to have references in regard to his reliability and also know whether his product had merit. We told him that this was no reflection on him or his product, but we had to do this in order to protect our readers.

He gladly gave us the references required, and we are investigating these now. If his product has the merit that he claims, his announcement will appear in our columns—otherwise it will not. And he has plenty of money to pay the advertising bills, too.

This manufacturer stated, "I have run up against something different here. Heretofore my life has been made almost a burden by advertising men soliciting my business. They do not seem to care whether my product appeals to their particular class of people or not. I am simply another victim—they want my business, that's all.

"As for investigating the truth of my claims, you are the first one to bring up the subject.

"You strike a new note—you say you will let me advertise provided I furnish references as to the worth of my product. Moreover, you do not talk space, but service. You offer to render a service—get up a definite, specific plan of presenting my product to my possible customers and give them information.

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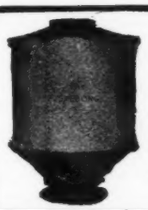

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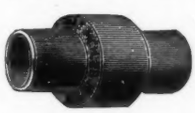

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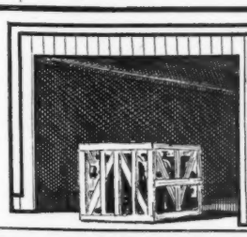

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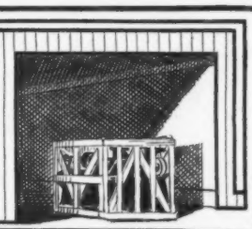

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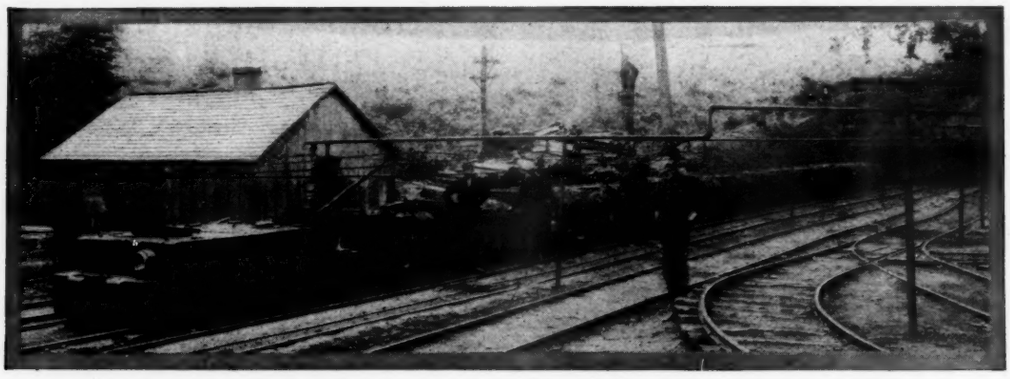

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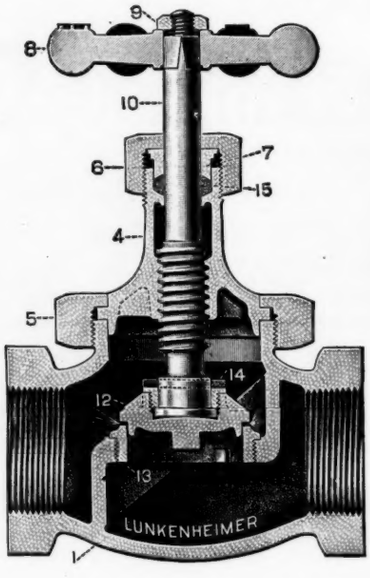

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